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"POLYTECHNICA" UNIVERSITY OF TIMISOARA
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Department of Mechanics and Vibration
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1st SESSION

NOISE

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THEORY OF ACOUSTIC METAMATERIALS: AN OVERVIEW

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Abstract - *In the paper the theoretical consideration of the acoustic metamaterials is given. Metamaterials, which are usually composite, are artificial materials whose properties differ from those observed in nature or in the constituent materials. Metamaterials which are suitable for acoustic wave absorption are presented. Acoustic absorber is a beam made of solid material connected with spring-mass subunits. The purpose of the subunits is to give a band gap where some frequencies of acoustic wave are stopped. Mathematical models for various types of connection of subunits in the metamaterial and absorber are discussed. Based on the analogy between electromagnetic and acoustic waves the concept of negative effective mass is introduced as a basic principle for modeling. Acoustic metamaterial beams based on one, two or multi-frequency vibration absorbers are discussed. Depending on connection of absorbers in the beam, the structure may absorb wave in one-direction (for example the longitudinal one) or waves in two directions (transversal and longitudinal). In the paper an overview of mathematical models and suggestions for further investigation are given.*

1. INTRODUCTION

Recently, various artificial structures such as acoustic metamaterials have been extensively studied to control the transmission of acoustic waves. Their potential application is in the field requiring the manipulation of group velocities of acoustic waves, for example, in acoustic filters, in precise spatial-spherical and microscopic control, but the main use is in noise reduction systems and acoustic cloaks. Namely, the 'conservative' noise absorbers have some limitation of noise reduction due to their dimensions and weight. It is shown that at a given frequency the level of sound transmitted through a partition will be reduced by 5-6 dB for every doubling of the mass of the partition, which implies that to improve the transmission loss by 30 dB the partition must become 32-64 times heavier.

To control, direct and manipulate sound waves acoustic metamaterials are designed. The idea for production of acoustic metamatirials issues from the analogy between sound and electromagnetic waves and the properties of mechanical and electromagnetic systems. Knowing that there is an analogy between of density and bulk modulus and electromagnetic parameters: permittivity and permeability, the realization of the project of acoustic material starts in 2000. Acoustic metamaterials have to be designed to achieve similar phenomena as electromagnetic metamaterials. As the electromagnetically induced transparency can control optical responses, the acoustic metamaterial is created based on a series of mechanical resonators arranged in a medium to realize slow sound wave at a frequency between the resonant frequencies of the resonators.

Investigation in metamaterials has two main directions: one, in forming the conception of new structures and the other, in designing new types of units which form the material. However, both are based on the concept of conventional vibration absorber.

2. CONCEPT OF VIBRATION ABSORBER

As it is well known, the conventional vibration absorber consists of a lumped mass m_2 attached with a linear spring k_2 to the mechanical system with mass m_1 excited with a harmonic force (see Fig.1). Equations of motion of the two mass system are

$$m_1\ddot{u}_1 + k_2(u_1 - u_2) = F_0 \exp(i\omega t), \qquad (1)$$

$$m_2\ddot{u}_2 + k_2(u_2 - u_1) = 0. \qquad (2)$$

where F_0 and ω are amplitude and frequency of the excitation force, while u_1 and u_2 are generalized coordinates. The solutions of equations are in general

$$u_1 = a_1 \exp(i\omega t), \quad u_2 = a_2 \exp(i\omega t) \qquad (3)$$

where amplitudes of vibration are

$$a_1 = \frac{F_0(k_2 - m_2\omega^2)}{(k_2 - m_1\omega^2)(k_2 - m_2\omega^2) - k_2^2}$$

$$a_2 = \frac{F_0 k_2}{(k_2 - m_1\omega^2)(k_2 - m_2\omega^2) - k_2^2} \qquad (4)$$

and $i=\sqrt{-1}$ is the imaginary unit. For this model only one local resonance frequency exists. The vibration absorber uses the 1:1 external resonance between the forcing frequency on the main system ω and the local resonance frequency of the absorber $\omega_2 = \sqrt{k_2/m_2}$ to transform the vibration energy to the absorber and stop the main system's motion (u_1=0).

Let us transform the two-mass system to a single-mass system with effective mass m_{eff} whose motion is the same as that of m_1. The effective mass is defined by treating this two degree-of-freedom system as a one degree-of-freedom one by assuming the internal absorber being unknown to the observer. In other words, the identity of the internal mass m_2

would be ignored and its effect would be absorbed by the introduction of an effective mass m_{eff}

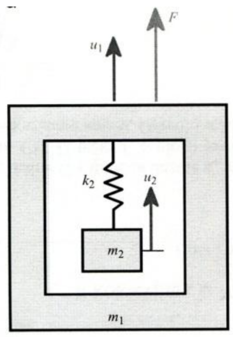

Fig. 1 *Mass-in-mass model.*

If the motion of the mass m_1 is u_1, the effective mass has also the motion u_1. The linear momentums for the both models have to be equal, i.e.,

$$m_{eff}\frac{du_1}{dt} = m_1\frac{du_1}{dt} + m_2\frac{du_2}{dt}. \qquad (5)$$

Substituting the assumed solution (3) it is

$$m_{eff}a_1 = m_1 a_1 + m_2 a_2. \qquad (6)$$

Motion of the mass m_2 is given with the equation (2). Substituting the assumed solutions (3) it is

$$-m_2\omega^2 a_2 + k_2(a_2 - a_1) = 0. \qquad (7)$$

After some modification the equations (6) and (7) yield the effective mass

$$m_{eff} = m_1 + m_2\frac{k_2}{k_2 - m_2\omega^2} \qquad (8)$$

which is for $\omega_2 = \sqrt{k_2/m_2}$

$$m_{eff} = m_1 + m_2\frac{\omega_2^2}{\omega_2^2 - \omega^2}. \qquad (9)$$

Analyzing the relation (9) it is obvious that the effective mass depends on the ration between the excitation frequency and natural frequency of the system

$$m_{eff} = m_1 + m_2\frac{1}{1 - \frac{\omega^2}{\omega_2^2}}. \qquad (10)$$

Three modes of motion are evident: 1) acoustic mode when $\omega<\omega_2$, 2) resonant mode when $\omega=\omega_2$ and 3) optic mode when $\omega>\omega_2$.
For the acoustic mode the effective mass is positive. In the resonant mode the effective mass is theoretically infinite. In the optical mode the effective mass is negative for

$$m_1 + m_2\frac{1}{1 - \frac{\omega^2}{\omega_2^2}} < 0. \qquad (11)$$

Otherwise, it is positive. For a parameter values which satisfy the condition (11) the relation for variation of the effective mass as the function of the frequency is plotted in Fig.2.
Differentiating the relation (5) we have

$$(m_{eff} - m_1)\ddot{u}_1 = m_2\ddot{u}_2\,. \qquad (12)$$

Substituting (12) into (2) we obtain

$$k_2(u_2 - u_1) = -(m_{eff} - m_1)\ddot{u}_1 \qquad (13)$$

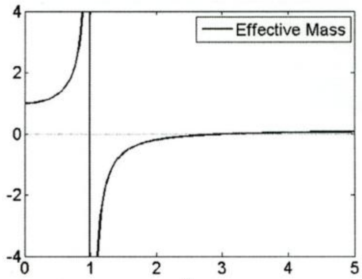

Fig. 2 *Dimensionless effective mass m_{eff}/m_1 as a function of ω/ω_2* [1].

Equation (1) and (13) give

$$-m_{eff}\ddot{u}_1 = F_0\exp(i\omega t) \qquad (14)$$

The effective mass is the ratio between the excitation force and acceleration of the mass m_1

$$m_{eff} = \frac{F}{\ddot{u}_1} = \frac{F_0}{-\omega^2 a_1} = -\frac{F}{\omega^2 u_1} \qquad (15)$$

If $\omega_2=\omega$ the effective mass tends to infinity. For that value the motion of mass m_1 is zero and the inertial force of the mass m_2 is equal to the excitation force: $F(t) = m_2\ddot{u}_2$. So, the external force is eliminated with the inertia force $-m_2\ddot{u}_2$ through the spring k_2. This is the concept of vibration absorbers.
From the previous consideration and the Fig.2 following is concluded:
If $\omega<\omega_2$ and the effective mass m_{eff} is positive in the acoustic mode, the motions u_1 and u_2 are in phase. If $\omega>\omega_2$ and the effective mass m_{eff} is positive or negative in the optical mode, the displacements u_1 and u_2 are 180^0 out of phase. Then, the absorber works efficiently in the optical mode against the external acting on the mass m_1. The excitation is absorbed with the inertial force.

3. METAMATERIAL STRUCTURES

Santillan and Bozhevolnyi [2,3] formed an acoustic metamaterial based on series of resonators described in the previous section. The resonators are arranged along a tube. This metamaterial realizes slow sound waves at a frequency between the resonant frequencies of detuned resonators. An improvement of the model is done by Li *et al*. [4]. They created an acoustic metamaterial on the basis of a simple and compact structure composed of solely one string of side pipes settled along a main wave guide other than asymmetric resonators. Using this metamaterial the group velocities of acoustic waves can be manipulated to simultaneously realize diverse group velocities in one structure as negative group velocities: fast and slow waves. Near the resonant frequency in the metamaterial negative phase time is obtained and fast and slow acoustic waves are achieved.

3. METAMATERIAL STRUCTURES

In [5] a type of composite is presented which displays localized sonic resonances at 350-2000 Hz with a microstructure size in the millimeter to centimeter range. The schematic structure of the composite is plotted in Fig.3. “A” denotes a lead solid particle in centimeter diameter, “B” denotes a silicon rubber layer and “C” denotes the matrix material of epoxy.

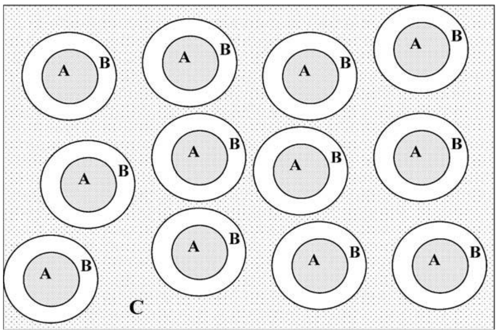

Fig. 3 *Schematic structure of the composite* [5].

Around the resonance frequencies the composite behaves as a metamaterial with effective negative elastic constraints and as a total wave reflector. When the microstructure is periodic this composite exhibit large elastic wave band gaps at the sonic frequency range.

Based on the theory given in [5], Sheng et al. [6] formed practically the composite with coated spheres which can exhibit resonance frequency in a fluid. The basic structure unit of locally resonant sonic material (LRSM) is a sphere core coated with a layer of silicon rubber (Fig.4).

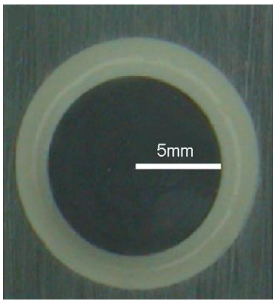

Fig. 4 *Basic structure unit of LRSM* [6].

As a resonator the core particle acts as the mass of the oscillator and the coating is a spring. If the fluid medium exhibits a resonant behavior at the same frequency as the coated spheres the acoustic absorption occurs. The question remains: How this can be realized?

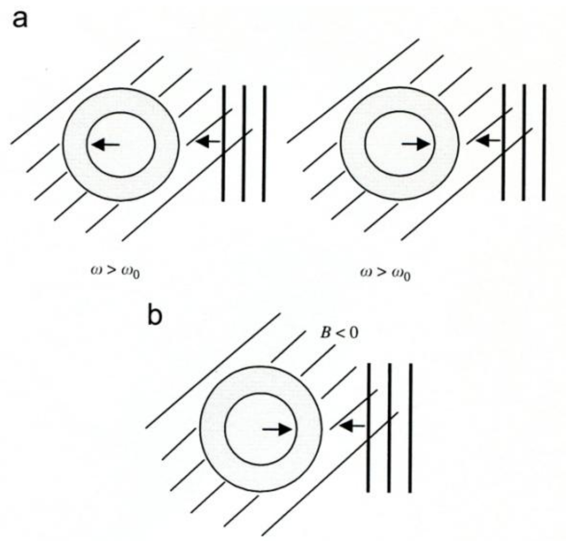

Fig. 5 *a) Schemes of relations between coated particles and medium, b) Scheme of acoustic metamaterial* [6].

A dense collection of structure units LRSM with the matrix material, having a lighter density than the core particle, realizes negative dynamic density within the resonance frequency range. When the core particle oscillates in-phase with the wave in the matrix medium ($\omega<\omega_2$), the dynamic mass density must be positive, as shown in Fig.5a left. When $\omega>\omega_2$ the core particle oscillates out of phase with the wave in the matrix medium as shown in Fig.5a right, the dynamic mass density can be negative, provided the density of the coated particles is sufficiently high. The core particles have a higher density than that of the matrix medium. Acoustic metamaterial is realized when the dynamic mass density is negative and addition the fluid medium is also negative with negative bulk modulus as a result (Fig.5b).

In the paper [7] the realization of a structure which responds as if it had negative mass to oscillations at a fixed frequency above resonance is suggested (Fig.6). The spherical lead core oscillates out of phase with the rigid, but light, surrounding shell. For this time harmonic motion the inertial force on the outside shell is in the same direction as the acceleration of the outside shell (but in the opposite direction to the acceleration of the ‘hiden’ lead core). Thus if the body is shaken at that frequency it will feel like it has negative mass. If the core was ellipsoidal in shape, or the rubber was replaced by a suitably anisotropic material, the effective mass of the body would be anisotropic.

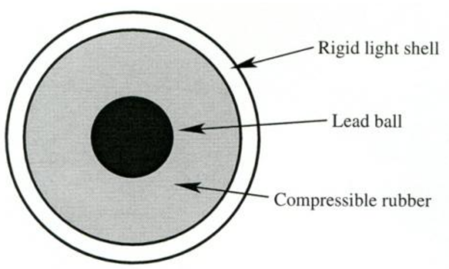

Fig. 6 *An element with negative effective mass* [7].

The suggested metamaterial made of elements given in [6] is experimentally tested in [8-10]. It was concluded that the density of metamaterial can be negative over a range of frequencies. Composites behave as anisotropic materials with complex density which depends on the frequency of oscillation. Milton and Willis [11] gave the theoretical consideration for dynamics of seemingly rigid bodies with composite structure. The system is modeled as one-dimensional (Fig.7) where n cylindrical cavities of length d are carved out from a beam of rigid material. In the center of each cavity is a lead sphere of mass m and radius r which is attached to the ends of the cavity with two springs each having the same spring constant K. Excitation is harmonical with frequency ω. For the resonant case it is $\omega^2=2K/m$. It is worth to say that the model is formed using the model of semiconductor where it is already known that the electrons and holes can have anisotropic effective masses due to their interaction with periodic potential. According to (8) the effective mass of the model in Fig.6, i.e., effective density for the model is

$$m_{eff} = M_0 + nm\frac{2K}{2K - m\omega^2} \tag{16}$$

where M_0 is mass of the rigid beam. The suggested model has the characteristic feature that its macroscopic properties depend on the resonant properties of substructures. Such resonant substructures give rise to new effects. In the models the motion of the rigid material violates Newton's law owing to the vibration of the internal masses: force equals mass times acceleration only applies if mass is replaced by effective mass which is non-local operator in time (a function of frequency under any purely harmonic excitation). In the model the macroscopic velocity is not the averaged velocity in composites with voids, because it is unclear what to take for the velocity in the void phase. There was a need to generalize the continuum elastodynamic equations which govern the effective response of bodies with or without voids. The weighted average of the local velocity is introduced as the macroscopic velocity where the weighting function is zero in the void phase on in 'hidden regions' which are not accessible to measurement.

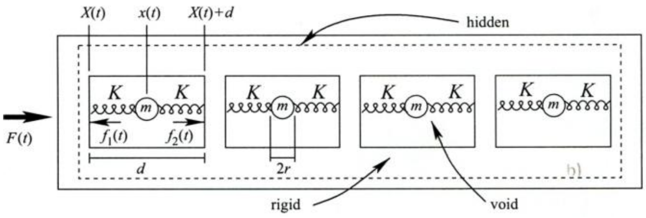

Fig.7 *A one-dimensional material where the effective mass depends on the frequency ω and can be negative* [11].

Instead of using springs one can fill each cavity with an elastically anisotropic (and possibly viscoelastic) material with the lead sphere being inserted in the center.

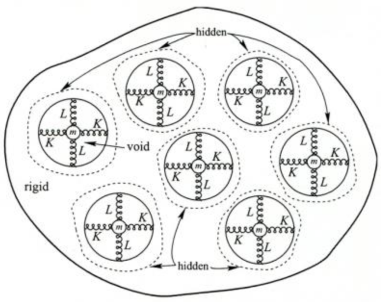

Fig. 8 *The two-dimensional model* [6].

In the paper [11] the model is extended to a two-dimensional one where the orthogonal springs in horizontal and vertical direction have the rigidity K and L (Fig.8). The dependence of the effective mass on frequency of the periodic force and the anisotropic property of the material is evident. The further extension is to a model with different unit cells (Fig.9).

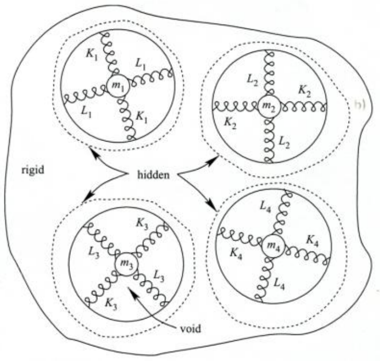

Fig. 9 *The generalized two-dimensional model* [6].

In the paper [7] a new unit cell model is suggested. The masses are approximated as point masses in the unit cell of the model of metamaterial (Fig.10). All springs which respond to elastic material have the same constant of rigidity.

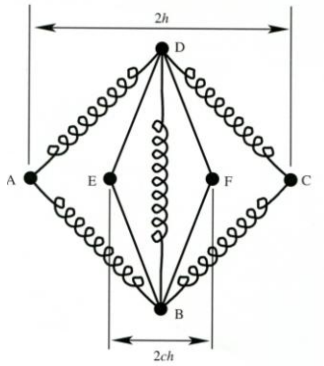

Fig. 10 *A unit cell of the metamaterial* [7].

The unit cell of a continuum model, which is conjectured that approximates the behavior of the discrete model, is plotted in Fig.11.

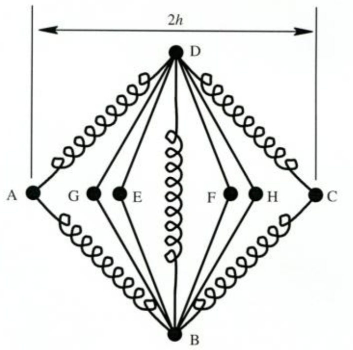

Fig. 11 *A unit cell of a continuum model* [7].

The black disks are heavy masses, and the remaining black areas are rigid but light material. The shaded areas are compressible material. The material surrounding the two black disks on the left side of the unit cell is sufficiently stiff that their motion is in phase with that of the surrounding rigid material, giving a positive effective mass. The material surrounding the two black disks on the right side of the unit cell is sufficiently compliant that their motion is out of phase with that of the surrounding rigid material, giving a negative effective mass.

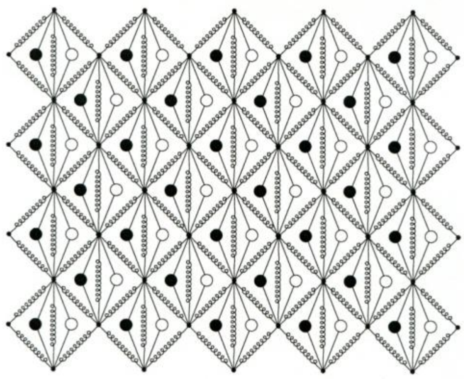

Fig. 12 *A suggested model of metamaterial* [7].

The triangular regions of material are highly compressible and are introduced so the junctions behave like hinges. The remaining area is void, or to make it a proper continuum, filled with a light highly compressible material. Unit cells are connected and form the model of the metamaterial (Fig.12).

The model of metamaterial is expected to have strange elastic behavior. The large solid black circles have positive mass, while the neighboring large white circles have negative mass at the given frequency. They are connected to the spring network by rods of fixed length, which alternatively can be regarded as springs with infinite spring constants.

In Fig13 the fabricated lattice metamaterial beam is shown.

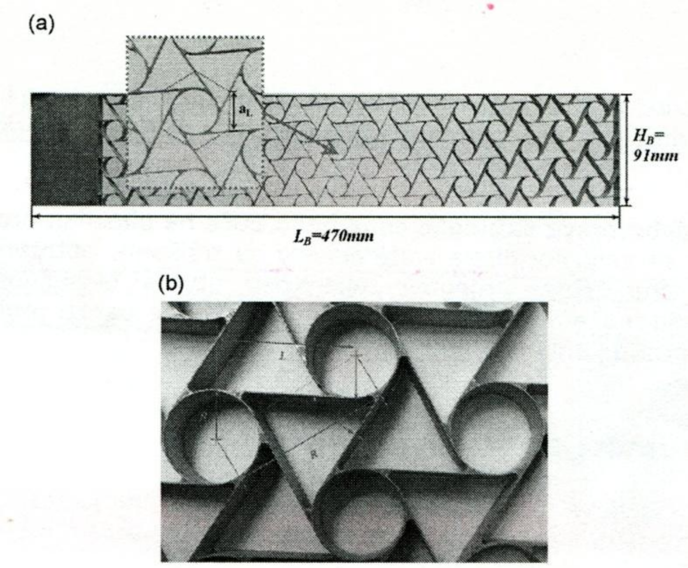

Fig. 13 *a) The fabricated chiral lattice beam and its zoomed unit cell and b) the topology of the hexagonal chiral lattice* [12].

The model is suitable to eliminate the propagation of the dispersive wave in the lattice for certain frequencies [12]. In Fig.14 the measured (open circles) and calculated (solid line) transmission coefficients as a function of frequency is plotted (Fig.14). The experimentally obtained values are in good agreement with calculated values. The result proves the theoretical consideration.

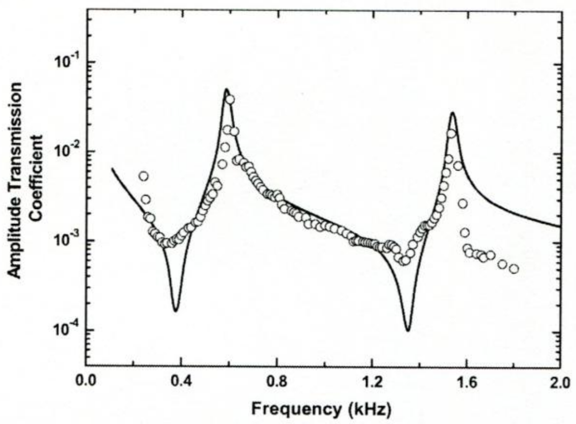

Fig. 14 *Measured (open circles) and calculated (solid line) transmission coefficients as a function of frequency* [5].

Finally, the structure in Fig.13 is non-homogenous. If it is required that the metamaterial behave like a homogeneous material described by its averaged material properties, its subunits must be much smaller than the shortest wavelength of waves propagating in it. The averaging may result in the existence of a useful but mysterious phononic stop-band that allows no waves within that frequencies range to propagate forward, and most current designs of acoustic metamaterials are based on the stop-band effect [11]. To manufacture such metamaterials with tiny subunits in order to have stop-bands, expensive manufacturing techniques are required including micro and nano manufacturing technologies.

4. CONCLUSION

Theoretically a significant number of model of acoustic metamaterials are developed. Their property is quite strange as they have negative effective mass i.e., negative effective density. One and two dimensional models are developed. Based on this concepts the three-dimensional models are also available. The experiments on some of the suggested artificial metamaterials are done. The results show their property to have a gap for certain frequency or frequency range which gives the possibility to be applied as acoustic absorbers. In future the presented metamaterial structures have to be realized and experimentally tested.

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REFERENCES

[1] H.H.Huang, C.T. Sun, G.I Huang, "On the negative effective mass density in acoustic metamaterials", *International Journal of Engineering Science* 47, pp. 610-617, 2009.

[2] A. Santillan and S.I. Bazhevolnyi, "V modes", *Physics Review B*, vol. 84, 064304, 2011.

[3] A. Santillan and S.I. Bazhevolnyi, "V modes", *Physics Review B*, vol. 89, 184301, 2014.

[4] X. Li, C, Xue, L. Fau, S. Zhang, Z. Chen, J. Ding, H. Zhang, "Simultaneous realization of negative group velocity, fast and slow acoustic waves in a metamaterial", *Applied Physics Letters*, vol. 108, 231904, 2016.

[5] P. Sheng, X.X. Zhang, Z. Liu, C.T. Chan, "Locally resonant sonic materials", *Physica B*, vol. 338, pp. 201-205, 2003.

[6] P. Sheng, J. Mei, Z. Liu, W. Wen, "Dynamic mass density and acoustic metamaterials", *Physica B*, vol. 394, pp. 256-261, 2007.

[7] G.W. Milton, "New metamaterials with macroscopic behavior outside that of continuum elastodynamics", *New Journal of Physics*, vol. 9, 359, 2007.

[8] Z. Liu, X. Zhang, Y. Mao, Y.Y. Zhu, Z. Yang, C.T. Chan, P. Sheng, "Locally resonant sonic materials", *Science*, vol. 289, pp. 1734-1736, 2000.

[9] Z. Liu, C.T. Chan, P. Sheng, "Analytic model of phononic crystals with local resonances", *Physical Review B*, vol. 71, 014103, 2005.

[10] A. Avila, G. Griso, B. Miara, "Bandes phononiques interdites en elasticite linearisee", *Physics Review E*, vol. 340, pp. 933-938, 2005.

[11] G.W. Milton, J.R. Willis, "On modifications of Newton’s second law and linear continuum elastodynamics", *Proceedings of the Royal Society A*, vol. 463, pp. 855-880, 2007.

[12] R. Zhu, X.N. Liu, G.K. Hu, C.T. Sun, G.L. Huang, "A chiral elastic metamaterial beam for broadband vibration suppression", *Journal of Sound and Vibration* 333, 2759-2773, 2014.

FREE FIELD ARRAY METHOD AS A TOOL FOR IDENTIFICATION OF CRACK NOISES IN HOUSEHOLD REFRIGERATORS

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Abstract - *This paper presents a new visualization method of complex sound sources in combined fridge freezers. Measurement method with sixty array microphones in free sound field conditions is used. Laboratory acoustic measurements using an algorithm of the complex sound sources visualization are performed. With this method, sound effects are successfully identified, localized and calculated. The individual crack noises emitted as a result of thermal dilatation of different types of material in the transitional cooling modes of the household refrigerator.*

1. INTRODUCTION

The first step to a successful noise reduction is the noise source identification (NSI). In this process, we can get assistance from the systems which visually place the sources of noise in the room [1].

If using one sensor (microphone or p-p probe) which measures sound pressure or sound intensity, we can determine the source of noise only by executing several measurements in different locations in the room and process the results numerically. When we deal with time-varying noise, the single sensor approach is not satisfactory.

In such case, a microphone array must be used for sound capture. All methods of processing the thus captured (acoustic) signals have been known among the professionals under the term 'acoustic holography'. The microphone array has been known as 'acoustic holographic camera'.

Refrigerators and freezers perform their function continuously, therefore it is important that their operation is as silent as possible and that they emit minimal noise. Their users perceive these features as on of the main criteria for the quality of appliances [9, 10]. Therefore, in addition to the declared sound power level of the appliances indicated on their energy label, the quality of the emitted noise is also important [11].

An additional issue is represented by the fact that refrigerators and freezers are becoming increasingly more complex. In addition to compressors, which have always been one of the main sources of noise, there is an increasingly frequent use of coolant directing valves, fans, air dampers, automatic ice makers and other components, the operation of which additionally contributes to the total noise level of refrigerators and freezers.

The noise level of refrigerators and freezers is also influenced by the type of the cooling system. One of the loudest is the so-called No Frost cooling system, which comprises the majority of the previously mentioned components. In such a cooling system, loud crack noises can also occur, which can be very annoying for the users, particularly during the night time. Although the crack noises occur in appliances made by different manufacturers, not much research in this field can be found in literature [12, 13].

Due to the short duration of sound impulses, conventional approaches and methods were not sufficient to solve these problems; we had to use a new method of visualization of complex sound sources. It is a measuring method using an array of sixty microphones in free sound field conditions. With this method, we successfully identified, localized and calculated the sound power level of the individual sound impulses, which helped us in determining the causes and searching for solutions for the elimination of the crack noises. At the same time, we used it for a quantitative comparison of the baseline and improved the configuration of the appliances, which is presented in more detail in the text below.

2. BUILDING BLOCKS OF THE ACOUSTIC HOLOGRAPHY SYSTEM

The system is composed of four basic building blocks:

- Non-uniform array of 60 microphones, 96 cm x 96 cm in size, with an optical camera in the center. The microphones are B&K Type 4957, the distance between the microphones ranges from 10 to 15 cm. A USB camera is installed - uEye model made by IDS Imaging Development Systems GmbH from Germany;
- A sampler: 5 modules B&K Type 3053-B-12/0 in the housing B&K Type 3660-C-000;
- Signal capture program B&K Pulse LabShop, version 20.0.0.455, and
- Signal processing program B&K Array Acoustics Post-processing, version 20.0.0.455.

The size of the used microphones (B&K 4957) is ¼ inch. Their frequency range is from 50 Hz to 10 kHz, dynamic range is from 32 to 134 dB. Their sensitivity is 11.2 mV/Pa. They have an integrated preamplifier which supports the TEDS protocol [3] (the microphone's characteristics are saved in the microphone and can be transferred to the sampler).

The measuring field with an optical camera in the center (Figure 1, left side) allows the results of measurements and calculations to be loaded on the image of the measured object. The sampler used (B&K 3053-B-12/0) supports 12 channels and has the frequency range up to 25.6 kHz. It is adapted for CCLD-microphones. The client communicates with it using the LAN-Xi protocol [4]. A system for 60 microphones requires 5 such modules installed in the housing B&K 3660-C-000 (Figure 1, right side. The latter also contains a suitable power supply unit and a LAN-Xi connection concentrator. The program B&K Array Acoustics Post-Processing is used for indirect analysis of captured signals. The basic version supports three methods of processing (licences for several other methods can be purchased additionally);

- delay and sum beamforming (DAS),
- SONAH, and
- wideband holography.

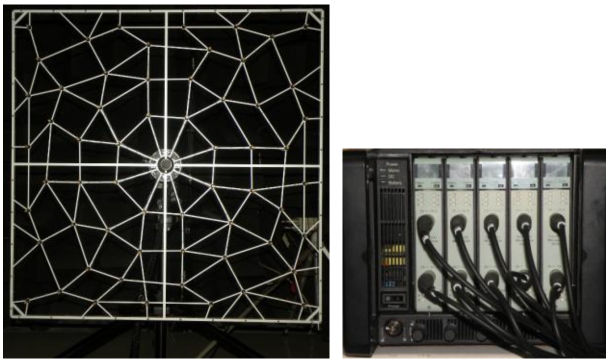

Fig1: *Brüel & Kjær system for acoustic holography, microphone array (left side), and sampler (right side)*

All methods calculate the sound pressure or the intensity for a virtual plane in space (*calculation plane*). All planes are parallel to the microphone array. For each measurement, any number of such planes can be set. The density of points on the calculation plane depends on the setting - the default distribution is a 2 cm grid.

The delay and sum beamforming method [5] is more accurate at higher frequencies. For each point on the selected plane, the delay of the sound to each microphone is calculated first, and then the appropriately delayed signals from all microphones are summed together. The calculation is repeated for all points on the monitored plane.

The SONAH method (*Statistically Optimized Nearfield Acoustic Holography*) [6] is based on the calculation of phase lag in the near acoustic field. During measuring, the microphone array must be in the immediate vicinity of the object being measured. The highest frequency that can be processed by this method depends on the maximum distance between the microphones in the array. For the array used, it amounts to 1 113 Hz.

The wideband holography method [7] combines good characteristics of both above-mentioned methods - it has a good resolution at low and high frequency, but is the most demanding in terms of calculation.

3. CRACK NOISES IN HOUSEHOLD REFRIGERATORS AND FREEZERS

Problems with cracking noise were occurring particularly in appliances with the No Frost cooling system. In domestic refrigeration and freezing appliances, the prevailing cooling systems are those with the compressor technology. Such cooling systems consist of at least the following closed-system components: the compressor which supplies work into the cooling system, the condenser which emits heat into the environment; the capillary tube which lowers the coolant pressure, and the evaporator which extracts heat from within the refrigerated space.

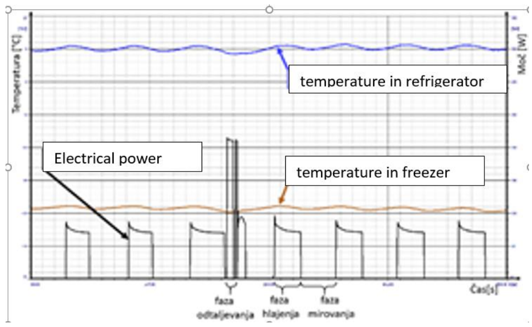

Fig. 2 *Temperature oscillation in refrigeration and freezing appliances*

A cooling system operates periodically; when the temperature inside the refrigerated space rises above a certain temperature, the cooling system turns on (start of the cooling phase), wherein the temperatures in the evaporator area sink below 0°C, which is followed by extraction of heat from refrigerated spaces by means of natural or forced convection. As soon as the temperature inside the refrigerated space sinks below a certain temperature, the cooling system turns off (start of the standstill phase). By alternating the cooling and standstill phases, the temperature in the refrigerated space oscillates around the set value, as shown in Figure 2.

Fig. 3 *Rime and clear ice on the evaporator*

During the cooling phases, moisture is released from the cooled air in the evaporator area and deposited in the form of rime and clear ice on the surfaces of the evaporator and the neighboring parts (Fig. 3). Accumulation of rime and clear ice reduces the efficiency of the cooling system; therefore, defrosting is required. In addition to manual defrosting, there are two automatic methods of evaporator defrosting:

1. Auto Defrost - passive method without heating devices in the evaporator area used mainly in refrigeration compartments,

2. No Frost - active method with additional heaters in the evaporator area, used mainly in freezing compartments and occasionally in refrigeration compartments.

In the Auto Defrost passive method, the temperature in the evaporator area rises above 0°C during each standstill phase of the cooling system, causing the rime and clear ice accumulated on the evaporator to melt and drain from the refrigerated space. In this defrosting method, the defrosting cycles take place during each standstill phase.

In cooling systems with the active No Frost defrosting method, the temperature in the evaporator area does not usually rise above 0°C during the standstill phases of the cooling system; therefore, in each subsequent cooling phase, an additional layer of rime and clear ice is deposited in the evaporator area. Defrosting takes place in provided defrosting cycles. During these cycles, heaters mounted on the evaporator are switched on and heat up the evaporator area. Each defrosting cycle lasts until the prescribed temperature above 0°C is reached in the evaporator area, which ensures that the accumulated rime and clear ice will melt and drain from the refrigerated space.

Fig. 4 *Frozen droplets on the cell*

At the end of each defrost cycle (in both passive and active defrost mode), water droplets of different sizes remain of the surfaces of the evaporator and surrounding parts; in the next working cycle of the cooling system, they freeze and turn into ice droplets. Figure 4 shows frozen droplets on the cell behind the vaporizer.

4. LOCALIZATION AND EVALUATION OF CRACK NOISE OCCURRENCE

The search for causes mostly took place in the semi-anechoic chamber of the Acoustics and Vibration Laboratory (Fig. 5), where we established the point of origin of the occurrences of crack noises and evaluated them.

Fig. 5 *Measurements by means of the SONAH technique*

For this purpose, we used a technique which is a combination of Nearfield Acoustic Holography (NAH) and Beamformimg (BF), and named Statistically Optimized Nearfield Holography – SONAH [14].

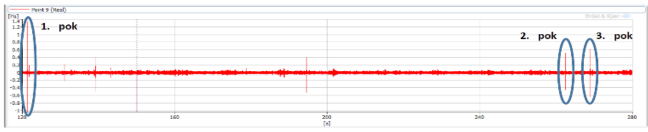

Fig. 6 *Time course of the sound emitted by the refrigerator in the first set of measurements*

Measurements of the emitted noise were carried out in two sets, i.e.:

- the first set - in the defrosting phase,
- the second set - after the defrosting phase, during the operation of the compressor.

In the first set, more crack noises were measured; an example of measurement is shown in Figure 6. In this case, we selected the three marked crack noises. Figure 7 shows their total sound power in the frequency range 125 Hz - 4000 Hz. Disregarding the first crack noise which shows non-characteristic behavior (the length of the signal is much longer that in all other cases, and the sound power is significantly lower than in the case of the second and third crack noise), we were able to establish that in both second and third crack noise, the majority of the source is in the location of the evaporator marked with number 2 in Figure 7.

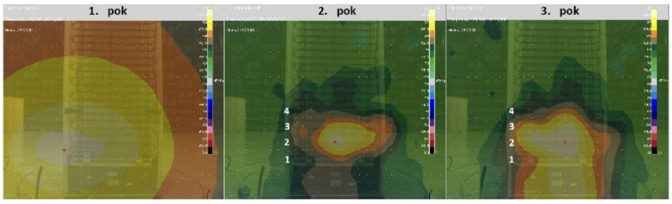

Fig. 7 *Total sound power of the three selected crack noises - sound images are not in scale*

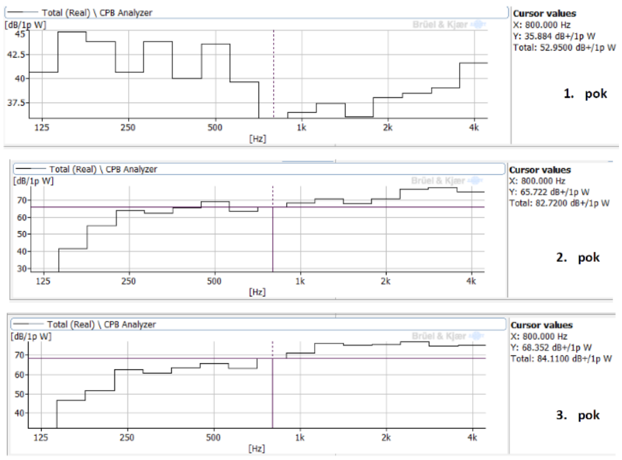

Fig. 8 *Frequency spectrum of the selected crack noises 125 Hz – 4 000 Hz*

The analysis of sound power by individual thirds (Fig. 8) showed that the total sound power in the case of the first crack noise is 53 dB, in the case of the second crack noise, 83 dB, and in the case of the third crack noise, 84 dB. In the frequency spectrum of the second and the third crack noise, there are no significant differences; the majority of the sound power above 70 dB was from the frequency range between 1 000 and 4 000 Hz.

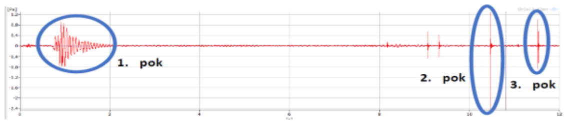

Fig. 9 *Time course of the second set of measurements*

In the second set, several crack noises were measured as well. For the analysis, we selected 3 crack noises (marked in Figure 9). Figure 10 shows the total sound power in the frequency range 125 Hz – 4 000 Hz for the example of all three crack noises from the second set of measurements. It is evident from this Figure that in the first crack noise, the main source of noise is in the area marked with numbers 2 and 3, and in the second and the third crack noise, the main source of noise had shifted to the right and upwards, so that the main source of noise was in the area marked with number 4. Both established sources of the main noise are in the area of the evaporator.

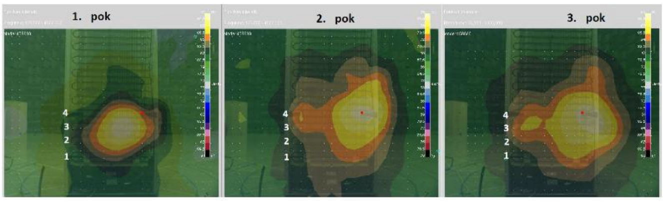

Fig. 10 *Total sound power of the three selected crack noises - sound images are not in scale*

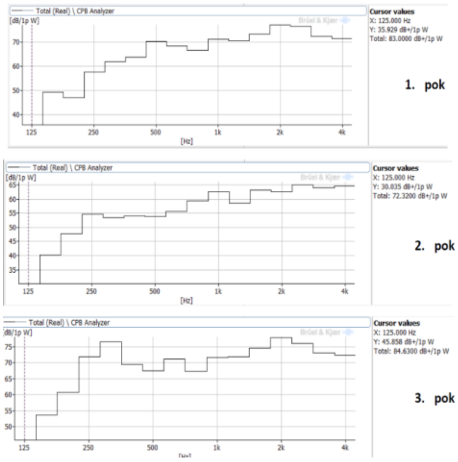

Fig. 11 *The frequency spectrum of the selected crack noises of the second set of measurements 125 Hz – 4000 Hz*

The analysis of sound power by individual thirds (Fig. 11) showed that the total sound power in the case of the first crack noise is 83 dB, in the case of the second crack noise, 72 dB, and in the case of the third crack noise, 84 dB. Similarly, as in the first set of measurements, the majority of the sound power, which is above 70 dB, is concentrated in the higher frequency range above 1 000 Hz.

Figure 12 shows the measurements of the total sound power (125 – 4000) for the first and the second set of measurements. The top three images show the measurements of the first set, and the bottom three images show the measurements of the second set of measurements. After comparing the results of measurements of both sets, we came to the conclusion that the maximum sound power in individual crack noises is comparable, i.e. approximately 83 dB. This result confirmed the fact that the crack noises are very loud and annoying. The difference occurring between the first and the second set is in the shift of the main source of the crack noise. In the case of the first set of measurements, the source appears on the position 2 (marked in Figure 10), while in the case of the second set of measurements, this source appears higher, at the position 3-4 (marked in Figure 12). Both established positions are in the area of the evaporator, as we already concluded in the first phase of searching for causes for occurrence of crack noises.

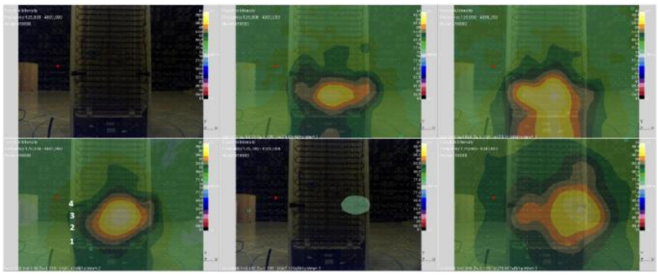

Fig. 12 *Comparison of total sound power of all six crack noises, top - measurements from the first set, bottom - measurements from the secont set. The figures are scaled to the highest measured sound power level of 84 dB.*

5. DESCRIPTION OF THE SOLUTION OF THE PROBLEM

5.1 Analysis of causes for crack noises

Considering the knowledge of the conditions around the evaporators of No Frost appliances during and after the defrost cycle, and based on the above listed findings, we set up two hypotheses about the causes for occurrence of crack noises. At the end of each defrost cycle, water droplets of different sizes remain of the surfaces of the evaporator and surrounding parts; in the next working cycle of the cooling system, they freeze and turn into ice droplets.

Crack noises can be heard when these water droplets are turning into ice. According to the first hypothesis, they occur when the evaporator is relatively tightly surrounded by neighboring parts (e.g. the cell and the air conveyor) from at least two opposite directions and when water droplets appear in the areas between the evaporator and at least one of the neighboring parts, but when turning into ice droplets, they push the evaporator away from the said neighboring part, If during this process, the tensions in the individual droplet exceed the burst tension of the ice, the latter bursts and vibrations in the form of an annoying crack noise occur.

According to the second hypothesis, the crack noises occur when the evaporator in is the vicinity of at least one part (e.g. the cell or the air conveyor) with a different temperature coefficient of length expansion, which can, upon changing of temperatures of the evaporator and the neighboring parts, and due to different thermal dilatations, cause tensions in ice droplets in the intermediate zone, which exceed the burst tension of the ice, causing the droplets to burst, resulting in vibrations in the form of annoying crack noises.

Hence, both hypotheses speak of the occurrence of vibrations due to bursting of ice droplets, which in themselves would not be so loud if the vibrations were not transferred to the insulated housing of the appliances. Due to relatively low density and high rigidity of the insulation of these housings, the vibrations are very well transferred and their sound effect can even be amplified.

Considering the above findings and hypotheses, we tested a series of versions of design improvements and operation algorithms, with which we tried to reduce the frequency and lower the loudness level of crack noises in refrigerators and freezers. Each of the improvements was evaluated by means of subjective identification, i.e. with the number and volume of crack noises during the monitored period.

5.2 Results of measurements of refrigerators' noise with structure modification

Considering the above findings and hypotheses, we tested a series of versions of design improvements and operation algorithms with which we tried to reduce the frequency and lower the loudness level of crack noises in refrigerators and freezers. Each of the improvements was evaluated by means of subjective identification, i.e. with the number and volume of crack noises during the monitored period.

A relatively simple procedure of placing aluminum foil between the evaporator and the evaporator cell proved to be the best solution.

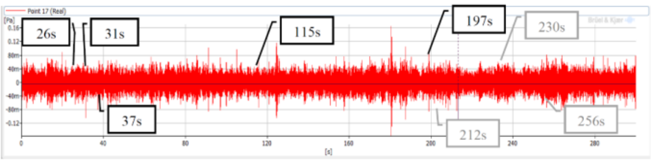

Fig. 13 *Time course of the final configuration in the defrost phase*

The same as in the case of the original configuration, the final solution was also evaluated by means of the combined SONAH technique. Figure 13 shows the sound pressure in the defrosting phase. Already from the comparison of time records of the sound pressure, we were able to establish that the sound pressure amplitude was decreased in case of all crack noises. Furthermore, the time of occurrence of individual crack noises could not be established on the basis of the sound recording; it could only be established by listening to the sound signal. Individual points in the time record, where the analysis of crack noises was carried out, are marked in Figure 13.

Here, it should be pointed out that the last three points were not analyzed, as the analysis of first points showed a minimum deviation, i.e. differences between individual crack noises.

The comparison of all five individual crack noises is shown in Table 1. As it is evident, the total sound power level of an individual crack noise does not exceed 44.6 dB (the latter occurs at 37 s).

Table 1 *Sound power levels of crack noises in the defrost phase*

Number of crack noises	Time of crack noises, s	Total sound power, dB
1	26	38.9
2	31	40.1
3	37	44.6
4	115	38.8
5	197	39.7

Figure 14 shows the locations of the source upon the occurrence of the crack noise for the example of all five analyzed crack noises. The sixth figure (right side, bottom) presents the analysis of operation of the refrigerator in the first 20 seconds of operation. All figures are scaled to the highest measured sound power level (44.6 dB).

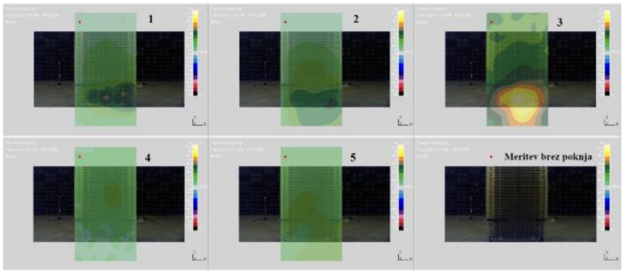

Fig. 14 *Comparison of total sound power level of all five crack noises in the defrost phase.*

The same procedure of evaluation of crack noises was used for the post-defrost phase, with the operation of the compressor (Figure 15). Similarly, the difference between the operation of the refrigerator and crack noises in this phase was so small that the occurrence of crack noises was impossible to determine from the time measurements of the sound pressure. In Figure 15, the individual crack noises are marked, but the last 5 crack noises were not analyzed due to the above-described causes. As evident from Table 2, the total sound power level does not exceed 41.14 dB (at 28 s).

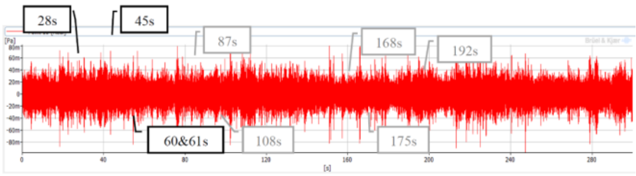

Fig. 15 *Time course of the final configuration after the defrost phase*

Figure 16 shows the locations of the source upon the occurrence of the crack noise for the example of all five analyzed examples. The fourth figure (right side) presents the analysis in the first 20 seconds of operation of the refrigerator. All figures are scaled to the highest measured sound power level (41.1 dB).

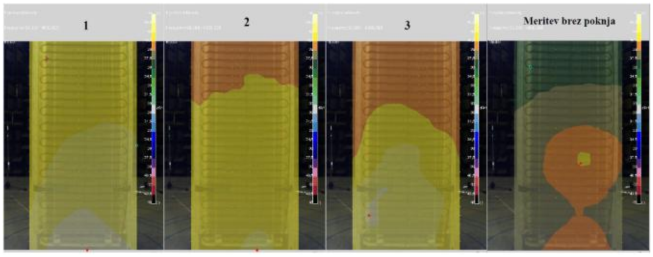

Fig. 16 *Comparison of total sound power level of all three crack noises after the defrost phase.*

Table 2 *Sound power levels of crack noises after the defrost phase*

Number of crack noises	Time of crack noises, s	Total sound power, dB
1	28	41.1
2	45	40.0
3	60&61	40.4

6. CONCLUSION

The main sources of noise in refrigerators and freezers, such as compressors, fans or circulation of coolant, can be identified relatively well and evaluated by using conventional measurement methods. We have been using them for years, both in development measurements, which help us in the search for the lowest possible noise level in the appliances, and in the execution of standard measurements, based on which the declarations of noise levels of appliances are determined. The noise from these sources has also been discussed in a number of scientific papers, where guidelines are provided which can be taken into consideration already in the phase of development of the appliances as well as in the process of eliminating problems on the market.

However, acoustic phenomena occurring during the operation of refrigerators and freezers are discussed and measured less frequently. This was also the reason for our presentation of an example of successful application of modern measuring equipment for the visualization of sound sources in transient phenomena which were reflected in the form of annoying crack noises. With its help, we managed to decrease the sound power level of crack noises from the original level of 84 dB to the final solution with 45 dB. By presenting this case, we are trying to encourage more frequent use of this equipment for more efficient and effective remedying of such deficiencies in the appliances.

Furthermore, we presented in the article the combination of the use of this equipment and the subjective evaluation of noise, and we tried to demonstrate the need for combining different approaches to efficiently and effectively solve such challenges. The subjective method of identification of less common sounds, e.g. in the form of monitoring the operation of appliances in offices and homes, and recording extraordinary events, must be used in the earliest possible phase of development of the appliances. By doing so, the majority of deficiencies can be identified in early phases rather than after launching the appliances to the market.

With the complementary use of various methods and equipment as well as the cooperation of both Gorenje's domestic and external experts, we can meet the users' expectations, and at the same time, with lower noise level declarations, we contribute to competitive advantages of our appliances.

REFERENCES

[1] Kevin Bernard Ginn, Karim Hadad: Noise source identification techniques: simple to advanced applications, bulletin of the conference Acoustics 2012, p. 1781-1786, 23 -27 April 2012, Nantes, France.

[2] PULSE Array-based Noise Source Identification solutions, http://www.bksv.com/doc/bp2144.pdf, 14 July 2016.

[3] Standard IEEE 1451.4-2004.

[4] Vladimir Hájek: Using LAN-Xi for Sound and Vibration Measurements, bulletin of the conference Acoustics High Tatras 2009, 28 - 30 September 2009, High Tatras, Slovakia.

[5] Jacob Juhl Christensen, Jorgen Hald: Beamforming, Brüel & Kjaer Technical Review, p. 1-48, No. 1, Year 2004.

[6] Jorgen Hald: Spatial Transformation of Sound Fields in the Automotive Industry, Brüel & Kjaer Technical Review, p. 1-24, No. 1, Year 1995.

[7] Jorgen Hald: Wideband acoustical holography, bulletin of the conference Inter-noise 2014, p. 44-57, 16 -19 November 2014, Melbourne, Australia.

[8] Jurij Munda, Damjan Zazula, Nikola Holeček: Delovanje sistema S-Cube za aktivno dušenje zvokov v pravokotnem akustičnem kanalu, (*Functioning of the S-Cube system for active sound damping in a rectangular acoustic channel*) bulletin of the conference ERK 2015, Vol. A, p. 315-318, 21 - 23 September 2015, Portorož, Slovenia.

[9] Jeon JY, You J, Chang HY. (2007) Sound radiation and sound quality characteristics of refrigerator noise in real living environments. Applied Acoustics 2007;68(10):1118–34.

[10] Jeon JY, Ryu JK. (2006) Allowable noise levels of residential noises. Inter-Noise, Honolulu, Hawaii, USA.

[11] Holeček N. (2015) Analiziranje in zmanjševanje hrupa gospodinjskih aparatov. (*Domestic appliance noise level analysis and reduction.*) Gorenje – information bulletin 1-3 / 2015, Velenje.

[12] Koruk H., Arisoy A. (2014) Identification of crack noises in household refrigerators. Applied Acoustics 89 (2015) 234–243

[13] http://www.dailymail.co.uk/sciencetech/article-2950678/Mystery-fridge-s-strange-NOISES-solved-Scientists-finally-trace-source-loud-sounds-haunt-kitchens-night.html

[14] Holeček N. (2016) Iskanje virov hrupa z uporabo akustičnih senzorjev in mikrofonske matrike. (*Search for sources of noise by means of acoustic sensors and microphone array.)* Gorenje – information bulletin 4-6 / 2016, Velenje.

[15] Bek M, Holeček N. (2015) Poročilo o preizkusu AH (*AH Test Report*) 0815.66-69.0002. Gorenje, Velenje

[16] Crocker J.W. (1998) Handbook of Acoustics. John Wiley & Sons, USA

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EXPERIMENTAL CHARACTERIZATION OF A PHOTO-ACOUSTIC MEASUREMENT SYSTEM

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Abstract – *Photoacoustic effect has increased application as a non-destructive method for characterization of thermal and optical properties of materials. Since the measurement of photoacoustic signal requires use of an electronic system, knowledge of the transfer function of the measurement system is a prerequisite for its application. This paper presents two different experimental techniques for the determination of the transfer function of a PA measurement system.*

1. INTRODUCTION

The PA effect is the generation of sound waves in a sample and its surroundings due to the exposure to modulated optical radiation [1]. The effect was discovered and reported by A.G. Bell at the end of the 19th century, but the proper explanation was given almost one hundred years later [1]. The explanation and further theoretical studies were based on the classical heat propagation theory, and they boosted experimental research and practical applications [2] of the PA effect.

A conceptual diagram of the PA measurement system is presented in Fig. 1. The light source emits light under control of the modulator. The modulated light beam irradiates the sample that absorbs a part of the light, and a part of the absorbed light is converted into heat.

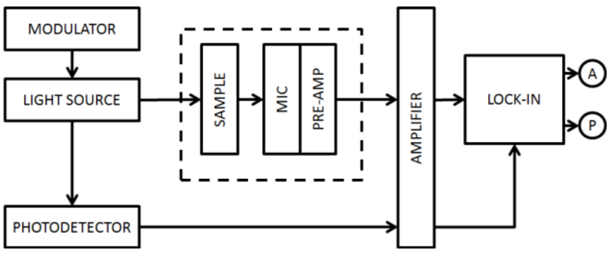


Fig. 1 *Photoacoustic measurement setup*

Heat transfer through the sample causes deformation of the sample and heat transfer to the surroundings causes expansion of the thin layer of the surrounding medium. Both the deformation of the sample and the expansion of the surrounding medium cause sound emission. The emitted sound waves propagate through a closed space of a PA cell and are detected by a microphone [3-4]. The majority of microphones used in the PA experiments are audio microphones with the frequency range 20 Hz-20 kHz. The signal of the microphone transducer is conditioned by the pre-amplifier of the microphone and led by a cable to an amplifier. The amplified microphone signal is led to one input of the lock-in detector. The amplified signal of the photo-detector that is illuminated by the same optical beam as the sample is led to the other input of the lock-in detector. The output of the lock-in detector consists of two signals: 1) the ratio between the amplitudes of the signals of the microphone and the photo-detector *A*, and 2) the phase difference between the signals *P*[4].

The described measurement system may be separated excitation part and detection part, where the excitation part consists of the light source, the modulator and the sample, while the detection part consists of the photodetector, the microphone, the amplifier and the lock-in detector. The excitation part has the variable light intensity $I(f)$ as the input, and the acoustic pressure $p_{PA}(f)$, which represents the PA response, as the output. The detection system has the acoustic pressure as the input, and if the two outputs of the lock-in detector at modulation frequency f, $A(f)$ and $P(f)$, are considered in the form $\underline{Y}(s=i2\pi f) = A(f)\cdot\exp(i2\pi fP)$, the Laplace transform of the output of the detection part and the complete PA measurement system may be represented as

$$\underline{Y}(s) = \underline{p}_{PA}(s)\underline{G}(s) \tag{1}$$

where $\underline{p}_{PA}(s)$ stands for the Laplace transform of PA response and $\underline{G}(s)$ stands for the transfer function of the detection part of the PA measurement system. If $\underline{G}(s)$ is known, then the amplitude $p_{PA}(f)$ and the phase $\varphi_{PA}(f)$ of the PA response may be extracted from the PA measurement data as

$$\underline{p}_{PA}(s) = \frac{\underline{Y}(s)}{\underline{G}(s)} \Rightarrow p_{PA}(f) = \frac{A(f)}{\left|\underline{G}(i2\pi f)\right|} \qquad \varphi_{PA}(f) = P(f) - \arg \underline{G}(i2\pi f) \tag{2}$$

Since the optical processes in the excitation part of the PA measurement system are much faster than the processes in the detection system, the transfer function of the complete PA measurement system (which has the modulated light as input and the outputs of the lock-in detector $A(f)$ and $P(f)$ as the ouputs), may be considered to be proportional to $\underline{G}(s)$. The transfer function of the PA measurement system may be determined using various techniques. The experimental characterization of a measurement system, also known as the calibration of the measurement system, comprises measurement of the output of a system that is exposed to a known input. Various calibration methods use different types of input signals. Acoustic systems are usually calibrated by harmonic input and white noise input.

The analysis given in the paper presents two different techniques for calibration of the PA system that is being developed at the Faculty of Mechanical and Civil Engineering in Kraljevo. The results obtained by the two techniques are compared, and the conclusion on their applicability and further directions of research are derived.

2. EXPERIMENT

Due to the small amplitudes of the PA generated sound, the distance between the sample and the microphone should be as small as possible. Microphones with small dimensions are the most suitable for the purpose, and in this research small commercial microphones manufactured by the company Kingstate Electronics Corp are studied, two KEEG1538WB-100LB microphones with 4 mm diameter (Fig. 2a) and one KECG2742TBL-A microphone with 6 mm diameter (Fig. 2b). The manufacturer declares the same frequency response for both types of microphone (Fig 2c.), which is flat in the range 20 Hz – 7 kHz. The power supply for the microphones is provided by a 9V battery and a LM78L voltage regulator, as presented by the section designated with dashed lines in Fig. 5.

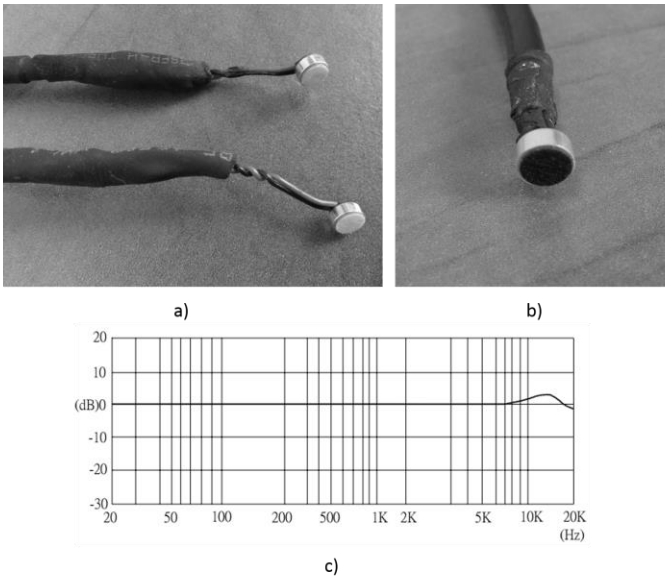


Fig. 2 *The microphones with 4 mm (a) and 6 mm (b) diameters and c) their frequency response*

The response of the PA measurement system was measured in the anechoic room of Faculty of Electronic Engineering, University of Niš, located in Svrljig. The excitation used for the calibration of the PA measurement was simultaneously measured using the reference acoustic measurement system that consists of the microphone B&K 4188-A-021 and the data acquisition system B&K 3560-B driven by the Time data recorder application of the Pulse software package. The aim of the measurement of the excitation signal is to take into account the influence of the frequency response of the sound sources that converted electric excitation signal into acoustic excitation. The same excitation setup and the same positions of the microphones were used in all measurements (Fig. 3).

The frequency response of the B&K 4188-A-021, shown in Fig. 4, is flat in the range 10 Hz – 10 kHz. However, the data acquisition system B&K 3560-B has the built-in high-frequency filter with cut-off frequency 22.4 Hz that reduced the passband of the reference measurement system. The sampling rate was set to be 65536 Hz, as it is the highest sampling rate supported by the Time data recorder application. The driver application stores data in the PTI format, but has built-in capability to convert the output data into MAT format used by the data processing software package Matlab.

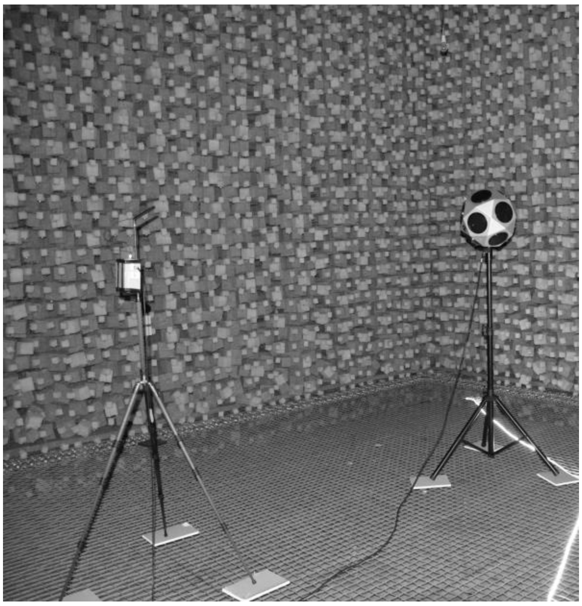


Fig. 3 *The sound source and the PA measurement system at the anechoic room during the experiment*

Two types of the excitation were used, resulting in two methodologies for characterization of the PA measurement system.

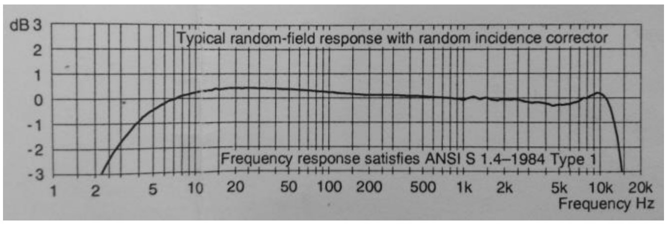


Fig. 4 *Frequency response of microphone of the reference measurement system*

2.1. Excitation by white noise

In theory, white noise is a random signal that has a frequency characteristic with constant amplitude, $|X_{WN}(f)| = C$, and random phase over all frequencies. Consequently, the amplitude spectrum of the output of a measurement system excited by white noise signal, $|Y_{WN}(f)|$, is proportional to amplitude of the transfer function of the measurement system, $|G(f)|$, since

$$\left|Y_{WN}(s)\right| = \left|G(s)\right|\left|X_{WN}(s)\right| = C \cdot \left|G(s)\right| \tag{3}$$

Therefore, the white noise signal is suitable excitation signal for experimental determination of amplitude of transfer function of measurement systems.

The white noise signal is generated using B&K Power Amplifier Type 2734 with embedded white noise signal generator, whose output was led to B&K Omnipower Sound Source Type 4296. According to the manufacturer specification, the white noise signal generated by the power amplifier has the flat amplitude of frequency characteristic in the frequency range 50 Hz – 5 kHz.

The output of the microphones was recorded using M-Audio Fast Track Pro USB digital sound card with the internal amplifier. The 24-bit resolution and 88200 Hz sampling rate were selected. The measurement was carried out using the Adobe Audition software package and the results were stored in WAV format.

For each of the microphones, seven measurement sessions were performed, each with duration of 16 s. The amplitude spectra of the microphone output signals, as well as the amplitude spectra of the outputs of the reference measurement system, recorded during the all seven sessions, were averaged. Finally, the amplitude spectra of the output of the reference measurement system (in dBs) were subtracted from the amplitude spectra of the microphone outputs (also in dBs), and the obtained result represents the transfer function amplitude of the PA measurement system.

2.2. Excitation by sweep signal

The sweep signal is the signal with constant amplitude and frequency that uniformly increases (sweeps) during the time [5]. In many practical applications, such as echolocation systems, sweep signals are used instead of impulses. With that inspiration, the idea presented in this paper is to use the sweep signal to determine the impulse response of the PA measurement system. If the spectrum of a measured system response to the sweep signal is $Y_S(f)$, then the impulse response of the measured system in frequency domain, $H(f)$ as well as in the time domain $h(t)$ may be determined using an appropriate digital filter function with frequency response $F(f)$ called the inverse filter and subsequent convolution [6]:

$$\begin{gathered} H(f) = F(f) \cdot Y_S(f) \\ h(t) = real\left\{iFFT\left[H(f)\right]\right\} \end{gathered} \tag{4}$$

The transfer function of the PA measurement system is Laplace transform of the impulse response $G(s) = \mathrm{L}[h(t)]$.

The sweep signal amplified by a commercial Sony TA-FE510R amplifier was radiated using the omnidirectional dodecahedron-shaped sound source of the Laboratory for Acoustics of the Faculty of Electronic Engineering, University of Niš. The exponential sweep signal with frequency range 20 Hz – 22050 Hz was recorded in digital WAV format as a file onto a computer. The excitation was reproduced using the 16-bit D/A card NI USB 6351 with sampling frequency 65536 Hz.

The amplifier of the PA measurement system is made using OP07 CP operational amplifier, as presented in Fig. 5. Using the resistor series, the gain of the amplifier may be selected to be 2, 4.9, 7.8, 11, 19, 48, 70 and 101. The bandwidth-gain product of the amplifier may be estimated to be close to the bandwidth-gain product of the OP07 CP operational amplifier, which is close to 0.5 MHz.

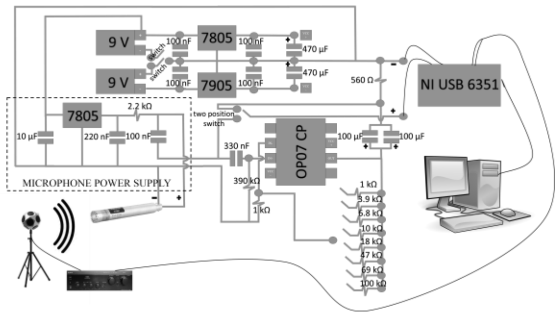


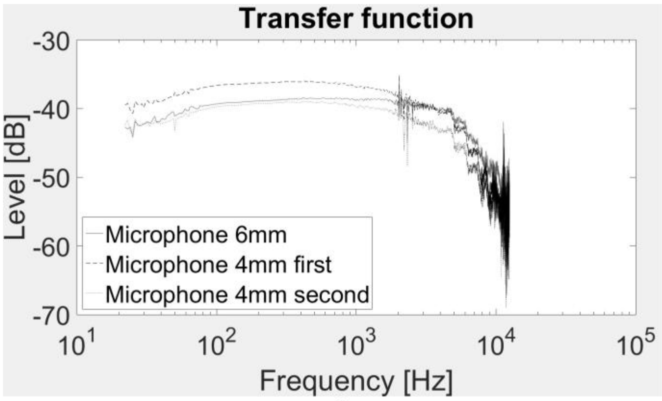
Fig. 5 *Experimental setup for measurement of the microphone impulse response*

The output of the amplifier is led to the NI USB 6351 data acquisition card that enables 16–bit A/D conversion.

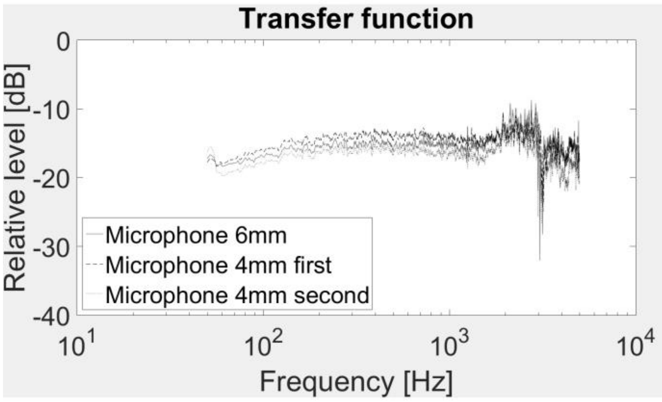
For each of the microphones, five measurement sessions were performed, each with duration of 16 s, as in case of white noise excitation. The amplitude spectra of the microphone output signals as well as the amplitude spectra of the outputs of the reference measurement system recorded during all five sessions were averaged. Finally, the amplitude spectra of the output of the reference measurement system (in dBs) were subtracted from the amplitude spectra of the microphone output (also in dBs), and the obtained result represents the transfer function amplitude of the PA measurement system.

3. RESULTS AND DISCUSSION

The transfer functions of the three studied PA measurement systems with different microphones are presented in Fig. 6.



a)



b)

Fig. 6 *The amplitude of the transfer function of the studied PA measurement systems using excitation by a) the white noise and b) the sweep signal*

The transfer function amplitude in the frequency range 50 Hz – 5 kHz determined using the white noise shows variation about ±10 dB in the range 2-3 kHz, and about ±20 dB close to 3 kHz, see Fig. 6a.

When the sweep signal was used to determine the transfer function in the frequency range 20 Hz – 22050 Hz, the variations about ±5 dB appear in the range 2-2.6 kHz, with several approximately equidistant peaks with 50 Hz difference. However, the large variations at the frequency of 3 kHz are not observed indicating that the feature may be an artifact of the experimental setup with white noise excitation and not an intrinsic characteristic of the PA measurement system. If the variations in the frequency range 2-2.6 kHz are also considered to be an artifact of the experimental setup with sweep excitation because they do not appear in the transfer function obtained by the white noise excitation, then the transfer function of the studied PA system may be considered flat (with variations less than ±3dB) in the frequency range 20 Hz – 6 kHz.

Since the gain of the amplifier used in the experimental setup for characterization by the sweep signal was set to be 70, it can be estimated that the cut-off frequency of the amplifier was around 6 kHz, which probably explains the drop in the amplitude of the transfer function of the PA measurement systems observed at frequencies higher than 6 kHz, which may be observed in Fig. 6b.

4. CONCLUSION

The paper presented the characterization of three PA measurement systems with the same concept and difference in the microphone only. Two methodologies for experimental determination of the transfer function of the PA system were used, the first based on excitation by the white noise and the second based on excitation by the sweep signal.

The results have shown that the obtained results differ at high frequencies, and there is an indication that each of the methodologies introduces its own artifacts into the obtained transfer function of the PA measurement system. Therefore, the application of both methodologies in the process of determination of the PA system transfer characteristic seems to be the most reliable approach.

The described process of development of the methodologies for calibration of the PA measurement systems shows that many variations in the experimental setups were made: sound sources, amplifiers and data acquisition systems used to implement the two methodologies were different. These variations represent additional sources of unreliability. The presented study can be used for comparison of different measurements approaches. In further research, uniform solutions and techniques will be used that will open the possibilities to compare the methodologies in a controlled manner, and to define the procedure for calibration of the PA measurement systems in a better way.

ACKNOWLEDGEMENT

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REFERENCES

[1] A. Rosencwaig, Photoacoustics and Photoacoustic Spectroscopy (R. E. Krieger Publishing Company, Malabar, Florida, 1980)

[2] Andrew C.Tam, "Applications of photoacoustic sensing techniques." Reviews of Modern Physics 58, no. 2 (1986): 381.

[3] Slobodan Todosijević, Slobodanka Galović, Jelena Tomić, Zlatan Šoškić, Developing Model of a Photoacoustic Measurement System, Proceedings of the 8th International Conference Heavy Machinery 2014, Zlatibor, 24-26.06.2014, pp. G 45-50, ISBN 978-86-82631-74-3

[4] Mihailo D Rabasovic, Marko G Nikolic, Miroslav D Dramicanin, Mladen Franko and Dragan D Markushev, Low-cost, portable photoacoustic setup for solid samples, Meas. Sci. Technol. 20 (2009) 095902 (6pp)

[5] Steven, W. Smith. "The scientist and engineer's guide to digital signal processing." California Technical Pub (1997).

[6] S. Müller, Measuring transfer-functions and impulse responses, in: Finn Jacobsen (Ed), Part I: Acoustic Signals and Systems, in:D. Havelock, S. Kuwano, M. Vorländer (Eds.), Handbook of Signal Processing in Acoustics, Volume 1, Springer Science, New York, 2008, pp. 65-86

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COMPARISON OF EFFICIENCY SILENCERS

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***Abstract** - For the purposes of this study were analyzed the results of several years of noise measurement testing shots in weapons, ammunition and silencers. Presented is a part of the results relating to silencers shot and give a conclusion on the reduction of noise levels depending on the distance and frequency. Those are other ways to reduce the noise level of the shot with a by pointing to their disadvantages*.

1. INTRODUCTION

The sound that is generated by the oscillation media can be periodic (tonnes), nonperiodic (noise) or aperiodic signal (attack). Sudden pressure change, which extends through the atmosphere attack occurs, which is called a shot when producing shoot a firearm, or the collision of powder gases and air. In the area around the extremely strong sound sources, due to the refraction of sound waves in the atmosphere, can occur silence zones, where sound waves move the upper atmosphere that are warmer at night, where they are reflected and fall to the ground. Therefore, it is possible to hear a gunshot to 20 km, but it is not heard at 10 km. The opposite phenomenon is the day when the lower layers of the atmosphere warmer than the upper.

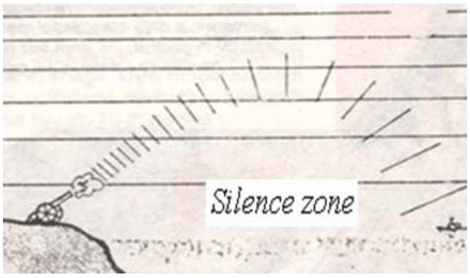

***Fig. 1** Silence zone in the area where the shot was not heard at night the mountain air is warmer, a gunshot is heard on, through*

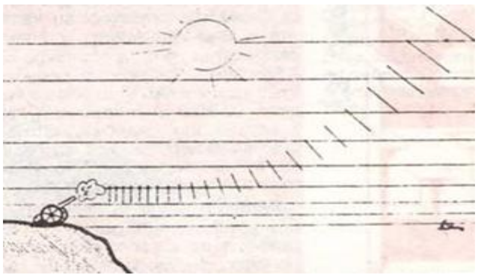

***Fig. 2** Refraction of sound waves during the day when the temperature is lower at altitudes*

Audio recordings of the shot can provide information about the location of weapons in relation to the microphone on the speed and trajectory of the projectile. The problem with the analysis of these diagram is echo (the overlap of direct and reflected signals) and diffraction from the surrounding barriers. For a complete understanding of the recorded signal the shot needs to know in addition to reflection and diffraction and other physical phenomena that accompany the propagation of sound: divergence, absorption, refraction, turbulence, etc.

Impulse noise of a gunshot is a complex acoustic effects arising from multiple sources, or in:

- Movement mechanism, metallic sound,
- Adiabatic expansion process in a barrel, burst at the muzzle, the primary, the dominant sound of the shot,
- Movement ballistic missiles, secondary shot
- impact missiles at the target.

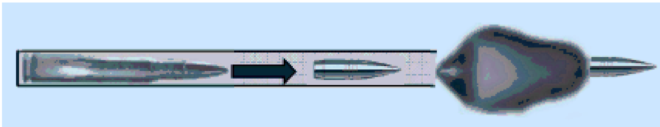

***Fig. 3** The movement of the bullet*

Mechanism mechanical noise is of value in the weapons with silencers. When you pull the trigger before the bullet is fired hear many moving, working parts. The sound is generated by striking a return spring when pulling the trigger, shot the firing pin and the movement of the shutter (the shutter is significantly more output with automatic weapons).

Using electrical firing system (used to modify the shotgun that animals would not be disturbed) noise is reduced, but the extra set of burdens weapons (do not mind the single finds firing). For operating noise bursts weapons work may be significant especially for weapons with automatic filling. Mechanical noise is mitigated construction of weapons, automatic braking work (manual locking) or eliminate the use of non-automatic weapons in special operations. Price eliminating this noise is a small effective range and trajectory of the grain with a high crown which is the main cause of failure in the case of moving targets and misjudgments distance.

Sonic boom on the muzzle, follows the sudden release of gases and large quantities of energy in the gun barrel after the explosion of gunpowder and chemical reactions of rapid transition of powder from a solid to a gaseous state. Sharp high-frequency sound (120 dB or more, depending on the caliber) products and detonation caps after the initial shock of the needle in the capsule, in the the chamber, but mostly absorbed into the arms somewhat reduced by the use of dampers. The combustion of gunpowder in tubes produce extremely expanding and increasing the volume of heated gas, or sharply increased pressure pushes the projectile forward (the ratio of the mass of the projectile and the thrust gas), which flies out of the tube and around which occurs fluctuation of air. Boiling gunpowder gases behind the projectile out of the tube under great pressure (for small arms, about 100 bar, 200 kg/cm^2) true three-dimensional shock wave causing oscillation compaction and air. The explosion of the mixture (created by mixing hot (for small arms around 1000 ºC) with cold gases oxygen from the atmosphere on the muzzle generates sound waves spherical shape, whose center is farther from the muzzle.

Fig. 4 *Firing of weapons*

Crack has a very short duration and high amplitude (eg. The whole acoustic effect can last for 3 ÷ 5 ms and peak sound pressure level is 150 ÷ 165 dB). At the source of the first positive pulse duration can be less than 0.5 ms, and for larger calibers milliseconds.

Although bang looks like a point source of sound, it does not radiate sound energy symmetrically. The weapon has a strict directivity characteristic of each weapon. The brake to influence the orientation of which is dependent on the frequency, lower frequencies are more higher than unidirectional. The noise level is the highest in the axis orientation of the tube and decreases with increasing angle of it.

Near the mouth of the tube, where the dynamic pressure of the shock wave near the static of 100 kPa (194 dB), the newly formed non-linear sound wave is a physical phenomenon. In these troubled airwaves changes the speed of sound + (densely) in - (vacuum). Close sources is faster attenuation of sound pressure level, which is the basis for the correction of non-linear in a linear model. At greater distances the linear propagation effects. The limit for the correction of non-linear model in a linear sound emission is given in ISO 17201 (154 dB), the process of NT Acou 099 (130 dB).

If the missile is faster than sound, this bang on the muzzle joins the supersonic bang.

The sound of bullets

Missile after thrown from the tube, if there is a speed greater than the speed of sound in air breaks the sound barrier, sound waves from overtaking is fired, hitting the air particles and causes oscillatory process. These oscillations of the speed of sound spread in the form of spherical wave and the progressive sound waves outside the tube is secondary weapons (ballistic) shot.

Around the projectile moving at supersonic speed is formed area of supersonic air flow and a shock wave (called "N" because of the shape of a wave, but vacuum pressure max). Time interval, "N" between the waves over and under the pressure depends on the size of the projectile.

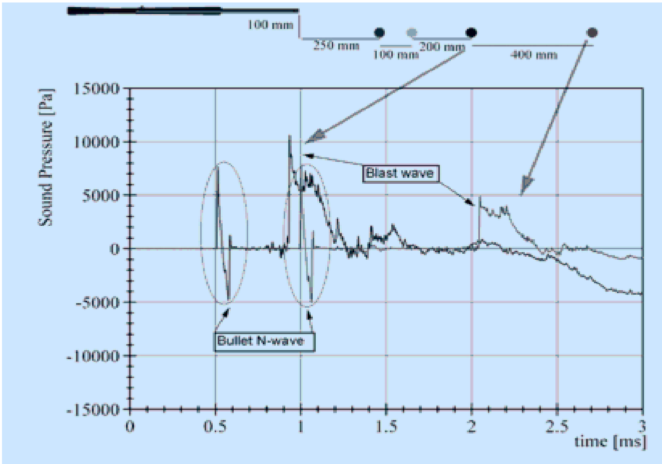

Fig. 5 *Signal shots*

The shock wave generated by the increasing speed of the projectile velocity flow about the projectile decreases, the pressure increases and the air flow separates from the projectile. When the velocity of the projectile is equal to the speed of sound shock wave is perpendicular to the movement of missiles and with increasing speed shock wave crossing the cone (Mach cone, Mach number M = v/c, where: c is the speed of sound, the velocity of the projectile) whose top, top missile and axle path of the missile.

All oscillations within the cone collide and mutually cancel each other, and only one in the area continue to expand and form a clearly articulated sound wave. This wave comes before the waves of shots on the muzzle, as an observer located in front of the first weapons hear the wave. The distance of the two waves changes depending on the angle of the viewer, increasing the angle decreases the distance, and at one point they run off into one shot. This is important with sound survey, because only on the basis of the shot on the muzzle can accurately determine the position of the shooter.

At a supersonic velocity missiles have clearly separate areas that make up part of the rarefied and compressed air environment, divided of the shock wave. On that line, which consists of a shock wave rapidly changing pressure and other parameters of flow. Compressed area is made of a series of spherical waves that creates Movement frontal part of the projectile. Angle (Mach angle) between the direction of movement of missiles and wave fronts, ie border diluted and compressed air environment depends on the speed of the projectile ($\boldsymbol{c = v \cdot \sin\alpha}$, Mach angle $\boldsymbol{\alpha = \arcsin(1/M)}$).

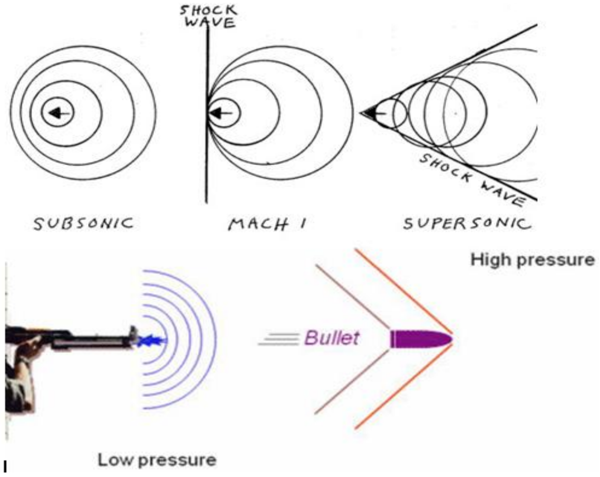

Fig. 6 *The shock wave shot*

Pressure in compressed field depends on the speed of the projectile and can reach 3 bar (300 kPa), which is three times more than normal atmospheric pressure.

Diluted zone under pressure behind the projectile attack air particles, and this swirling motion products zing (hiss) missiles. If a better aerodynamic shape of the projectile (cones on both sides) Whiz is weaker and vice versa. The missile is in flight behave like a top, fighting with the force of air resistance, so a wasp missiles trembles on its axis, which produces sound.

Is it possible to hear the sound of missiles that fly in a timely to get off?

Missile 7,62 mm (weight 8 g), which flies out of the pipe is 756 m/s, after one second yarn about 500 m and then his speed is reduced to 340 m/s (speed of sound). For two seconds the missile exceeds 800 m, and the speed of his fall to 260 m/s. Up to 500 m can not hear the zing before passing a firing missiles, then the pre hear the zing because the missile is slower than the sound. At a distance of 500 m can be heard whiz before the arrival of the projectile, but it is a bit of a second, but the audibility of a snarling projectile can not be used for timely protection.

In large caliber mortars 81-82 mm, initial velocity fired mine up to 250 m/s, with the "first shot" (a measure of your powder) initial velocity is about 130 m/s. Heard the first shot firing mortars and then whiz mine coming. Here and shot and projectiles whiz (mine) can be a warning. Flight time goes by 81-82 mm at 1000 m was about 24 seconds, and 2000 m flown in 29 seconds.

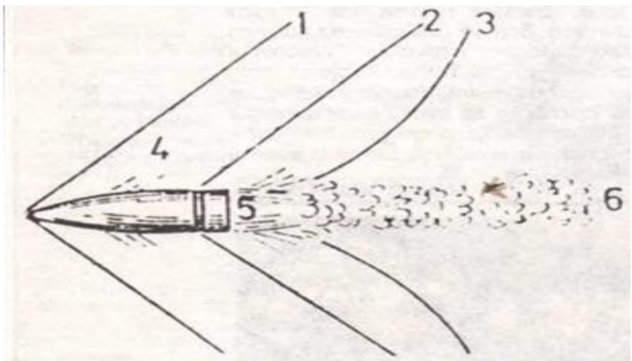

Fig. 7 *Flow around a supersonic missiles 1. shock wave, 2. wave of missiles groove, 3) the last wave, 4) Wave from the unevenness missile, 5. zone overpressure, 6) swirl*

While the basic shot frequency of about 500 Hz, ballistic (Mach), which occurs when breaking the sound barrier is 5000 Hz.

Target noise

On impact of the projectile at the target generated sound, which has no significant impact on the noise from artillery guns, or bombs, rockets and other missiles exploding is the dominant part of the noise. For percussive noise the same principles as the bang on the muzzle, or measurements made at greater distances and accuracy of results is not like to crack. For most of the explosion is assumed spherical, the same all-round sound waves.

Reduction of noise and silencers

The primary part of the shot powder gases and caps successfully reduced the use of silencers, where gases spread before meeting with the atmosphere, cooled to a temperature that prevents the occurrence of flames and flashes, slow down before going out so much that comes to the formation of a shock wave.

Secondary audio can be reduced by reducing the speed of the grain subsonic speed, which can be achieved by reducing or increasing your powder grain weight. Frequently used both methods combined. Products and ammunition subsonic speeds suitable for unconventional operations.

During the construction of silencers and ammunition adjustment, the point is not to just quietly fired shots, but the shots reduced the characteristics necessary to maintain order.

For silencers were first interested hunters on them in case of failure a chance to try again because the silent weapon would not be frightened deer.

The first silencer is patented in 1908 Hiram Maxim (son constructor machine guns), who saw a bang (shot) firearms causes a sudden release and spread of hot powder gases from the tube after the release of the projectile. He made a steel extension tube, wider and longer than the tube, and it tin barrier with slits, which were spun. Gases were before leaving the tube expand and cool in the silencer and slow vortex motion, cooled slowly stood out from the gun barrel after leaving the projectile.

The coefficient of expansion of propellant gases in relation to the volume of powder is very large. This information is important in the budget volume silencer, which should be at least 20 times greater than small arms to be effective silencer.

The degree of noise attenuation shots depends on the length of the tube silencer, its volume and the size of the opening through which the grain.

The muffler should be strictly in the axis of the tube that has a high heat capacity, but also high diffusivity of gas to be transmitted quickly, with openings for the passage of grains as similar as possible caliber weapons.

The efficiency of the silencer causes a number of factors, primarily the quality and construction of weapons (manual locking eliminates mechanical noise by firing bullets). Build quality reduce the gaps through which gases out that it is impossible to mute. Therefore, the attenuation of the revolver is not possible because of a gas between the drum and pipes (except some special version).

By function silencers are divided into silencers and suppressors, sluggers burst. Silencers have a purpose for subsonic ammunition (or supersonic, where the speed of the missile in the gunbarrel reduced to subsonic) while busters damped supersonic missile.

According to the structure, the external suppressors, which can be (de) mounted on arms and which are an integral part of weapons. Most of the weapons with silencers was originally a weapon, according to which self-adjusts suppressor, it is often the other way around to put extended threaded pipe, which disturbs the ballistic characteristics of the gun.

Integral suppressors are more efficient, have greater capacity for the same length of weapons and reduce the initial speed of the projectile, as part of drains powder gases from the barrel around which are mounted, through holes on it. This is the principle of absorption where the barrel is drilled along the entire length, and around it is the body of the silencer, filled with material that slows and cools the gases (brass wire, steel wool, etc.).

The most common is the principle of chamber where the body suppressors divided compartments with an opening for passage of the missile.

Inside the suppressors is a set of chambers of different shapes of which is usually the first expansion remained a vortex cooling gas. Chamber accepted rectilinear shock wave released propellant gas, turning it, breaking and swirling gases which force neutralizes itself. Whirling in a series of chamber damper slows down the movement of gas. Extended stay and more contact surface of the body and the silencer powder gases, slows and cools the powder gas. Upon exiting the machine, the speed of powder gases is significantly below the speed of sound and the conditions for the emergence of the shot noise are minimized.

Models with silencers funnel segments proved to be a good solution. The first two funnel segment (bearing sleeve and partition element) are exposed to the maximum pressures and temperatures, and are made of steel, and the rest of duralumin. Silencers with minted independent chambers have a successful reduction in noise by nonsuppressed weapons.

They are frequently found in which the suppressors structure is also applied over the aforementioned principles, or some other less efficient and well-known.

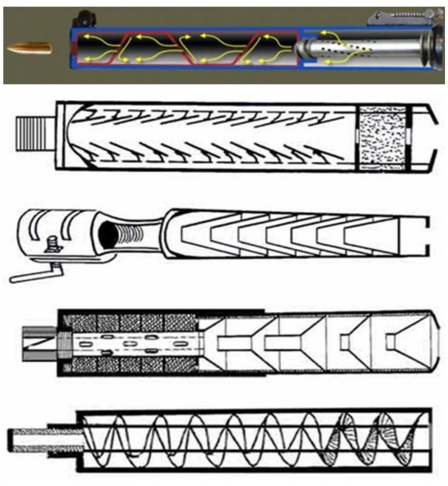

Fig. 8 *Components and various mounting silencers*

When supersonic missile firing a shot always hear due to sonic boom, but also in these weapons, the elimination scratch powder gases, noise is reduced and disappears when the velocity falls below the speed of sound. Silencers for such weapons were actually breakers burst. In their use due to constant refraction of sound through the environment, the distance from the shooter more than 200 meters is not possible to determine the direction from which the shot is coming, which extends their use particularly in long-range weapons.

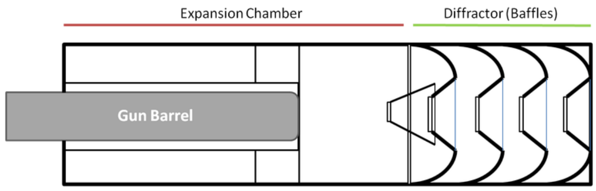

Fig. 9 *Suppressor inside*

Besides the thugs burst, gas brake (suppressor) (cost solution constructor Seberger Fil-a, for the caliber 5.56 mm and 7.62 mm; even the 12.7 mm) successfully dampens shot, transforming sound energy into heat. Only the resulting sound waves through the router and the chamber ricochet suggest themselves defying the heat releasing, so that after a few shots damper temperature much higher than the temperature of the tube weapon. The reduction of the noise level was so successful that the firing very little heard outside of weapons (such as the collision of billiard balls, and subsonic ammunition for even less). In support of the gas and brake, go to that precision is not significantly impaired, that the initial speed increased and that in dark conditions no flame in the firing, which hides the shooter.

Due to the needs of special units for efficient and quieter as weapons, a large number of models of weapons with silencers. In addition to the dim and very effective weapons developed is completely noiseless shots, in which the projectile leaves the barrel, while the gunpowder gases remain trapped in the sleeve piston pushes the grain.

The advantages of use of silencers

Can be used to burst weapons that are almost supersonic such silencers that reduce the rate below the bursts of sound. This reduces the pressure on missile reducing the speed at the exit of the pipe.

The silencer not only reduces the volume but also the emergence of flames from the tube for a long time because the expansion has already digested gas. This makes it difficult place to locate the shooter.

Using dimmer and less pressure gas leaving the tube means less recoil force that is transferred to the weapon, or the shooter, thus preventing his injuries.

Added mass (using silencers) and a smaller restoring force jerks can help with the precision bursts from automatic weapons because less movement is targeting.

The advantage of using silencer except in covert operations at military training, treatment of animals and the like.

When working with military training weapons can lead to permanent loss of hearing, if you are not using good quality hearing protectors, earrings. Even exposure to a single gunshot can cause hearing loss. Silencers may extend the operating time with weapons, reduce the fear of inexperienced shooters, protect hearing or reduce the need for the earrings, which enables more efficient communication and training between instructors and students.

Silencers are useful for veterinary purposes for a short time to treat animals. The shot in animals causes a dramatic stress and escape, which entails other environmental hazards.

Disadvantages of the use of silencers

Adding damper may unbalance weapons (moves to the center of mass), increases in the use of any discomfort, weight, and transport due to higher dimensions.

Mounting the silencers on the barrel affects the poor precision, the error is determined by firing with and without dampers.

Larger volume chamber silencers to reduce gas pressure and volume. Weapons must be easy to carry and easy to use, which limits the size and weight of silencers.

The greater the number and diameter of the holes makes it easier to leave the gas from the tube, or integral silencers to reduce the length of the tube to thereby speed projectile. The aim is that the size and arrangement of the holes do not disrupt the performance of the main weapons.

Using multi-chamber silencers is limited due to the deposition of unburnt gunpowder, if not used, and subsonic bullet.

2. TESTING OF NOISE REDUCTION GUNSHOTS

Mufflers can be said to have been successful in its purpose if in the measurement of sound pressure level at a distance of 5 m from the barrel and 45% in relation to the directrix, generate damping as in the table 1.

Table 1 *Criteria for evaluation of noise reduction shots*

Weapons	Damping dB(AI)	
	Successfully	Very successfully
Pistols	20÷25	25÷35
Automatic weapons	20÷27	27÷37
Shotguns, rifles	15÷18	18÷25

The efficiency of the damping shots is expressed by the difference in the level of noise generated by a real weapon for shooting with a silencer and without it (measured from the same place).

Numerous measurements of noise shots from small arms with and without silencers were done during various functional tests on the test site.

2.1 Analysis of test results

Part of the results of measurements of noise reduction shots from four time periods tests of different tools is provided in the following tables and analysis.

Efficiency by silencers domestic production (from the previous period) was demonstrated in the examination of various types of weapons, is given in the table 2.

Table 2 *Levels of attenuation the shot with homemade silencers*

Weapons	Attenuation to 2 m/45^0 from the mouth of the tube [dB]	Attenuation to 10 m/45^0 from the mouth of the tube [dB]
AR 7.65 mm M84	19,6	22,4
RA 5.56 mm M80	25,7	11,8
RA 7.62 mm M70	16.9	7,7
RA 7.62 mm M70 with a silencer shot	21,88	16
HRS 7.9 mm M76	18,4	9,6

Obviously, the best effect is achieved by damping the automatic pistol 7.65 mm M84 (10 m from the mouth of the tube 2-3 times larger than the others), which is logical because he has a subsonic bullet speed at the muzzle (about 315 m / s). It was also noted that the effect of damping in arms with the supersonic speed of grain on the muzzle (PPS 7.9 mm M76, M70 PA 7.62 mm and 5.56 mm M80 PA) near the biggest weapons that decreases with distance from weapons and that would disappear already at about 100 m from the muzzle.

Domestic company has until recently produced silencers shot is put on standard - neprigušivačka weapons that, in most cases with a silencer, supersonic speed with grains ($v_0 \geq 340$ m/s) at the muzzle. Under these conditions, a silencer "eliminate" a component of shots produced by the shock wave and flash powder gases to muzzle - two phenomena that make it easy to locate the source of the shot. Grain leaving the gun at supersonic speed, the products ballistic shot, which is a core component of the noise in the shooting and can not be stifled silencer.

In most cases, when the muzzle "no silencer weapons" put a damper shots caused a shift of the mean impact in relation to the median shot that weapon has a damper, often out of acceptable limits. On this anomaly, provided that the silencer is made out to be the most affected by the external accuracy of parts of the route around the muzzle of weapons that are used for bonding with a silencer (accuracy of workmanship and Alignment with the axis of the pipe thread to set compensator, external routing hide the flame front tombstones, etc.).

This means that before installing mufflers on a weapon must first determine the "suitability of weapons to set silencers". This is the reason why the same damper on an automatic rifle does not significantly affect its accuracy and on the other is unacceptable moves beyond a secondary grains touch and damage its elements.

Automatic Rifle 7.62 mm M70 with local special silencer shot (7.62 mm H39 with M78 grain weight 11.8 grams) and a silencer achieves effects silencer weapons - the muzzle velocity is about 285 m/s and there is no ballistic shot, a suppressor "eliminates" noise produced by the shock wave and flash powder gases to muzzle.

The price for this silence is a small effective range, changed the trajectory of the bullet, irregular guns, frequent failures in the case of moving targets and misjudgments distance.

Part of the analysis of noise measurements shots (second cycle) M21 rifle 5.56 mm in order to compare the tested damper (called 1 and 2) is given in Table 3 and Figures 10 and 11.

Table 3 *Comparison of the noise of the shot from a rifle M21 5.56 mm with silencers 1 and 2 and without silencers, measuring point the microphone at 5 m / 45º from the muzzle*

Rifle M21 5.56 mm with silencer 1	**117.1 dBA**(I), (5 kHz)
Rifle M21 5.56 mm with silencer 2	**115.6 dBA**(I), (5 kHz)
Rifle M21 5.56 mm without silencers	**127 dBA**(I), (800 Hz)

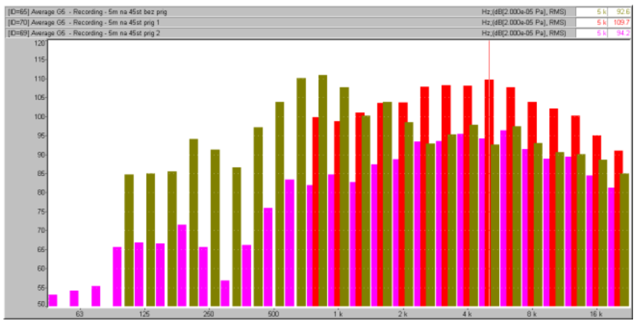

Fig. 10 *Spectrum shooting from rifles M21 5.56 mm with the damper 1, 2, and without silencer (5 m/45º)*

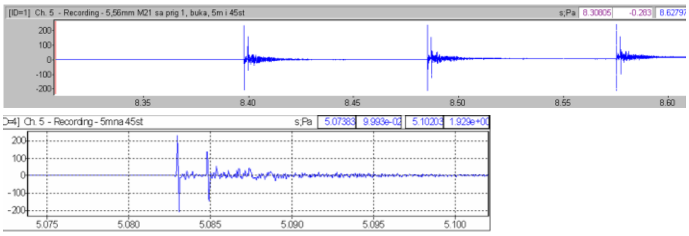

Fig. 11 *Rafal and individually gunshot M21 5.56 mm at the 5 m / 45 ° of the barrel*

The results of measurements of noise shots (third cycle) of 5.56 mm M21 rifle with a smaller propellant charge in order to compare the tested silencers (called 1 and 2) is given in the table 4.

Table 4 *Comparison of the noise a gunshot rifle M21 5:56 mm with a silencer 1, 2 and without silencers depending on the distance*

Measuring point: Microphone / line	With silencer 1 noise level (dBA(I))	With silencer 2 noise level (dBA(I))	Without silencers max noise level (dB(A(I)))
20 cm/45º	**127.2**	**135.0**	145
1 m/45º	**123.8**	**125.3**	135
2 m/45º	**120.2**	**122.3**	132
5 m/45º	**117.5**	**117.1**	**127**

In the next (fourth) cycle measurement objective was to select the optimal combination between one type of silencers and four guns, namely: two rifle SIG 516, 5:56 mm of various types and two SIG 716, 7.62 mm also different guns. Conclusion The analysis of the results of measurements of noise shots from the rifles of those with and without silencers, would be as follows:

- for the first rifle SIG 516, 5:56 mm successfully attenuation (above 20 dB (A)) was achieved in the range of 250 ÷ 1000 Hz for a shot only child, for a burst of 200 to 1250 Hz;
- for other rifle SIG 516, 5:56 mm successfully attenuation is in the range of 250 ÷ 1250 Hz for a shot only child, for a burst of 200 to 1000 Hz;
- SIG 716 rifle, 7.62 mm, successfully reduced is only in the range 500 ÷ 630 Hz for the only child shot.

It is obvious that the silencer with a rifle SIG 516, 5:56 mm, achieves better noise reduction.

In this cycle of measurement is measured, and the noise of moving parts of the mechanism of each listed rifle with a silencer, as illustrated in Figures 12 i 13.

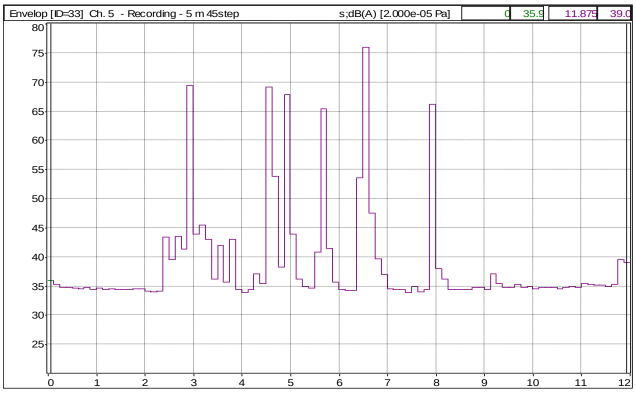

Fig. 12 *The noise mechanism SIG 516 rifles 5.56 mm with silencer*

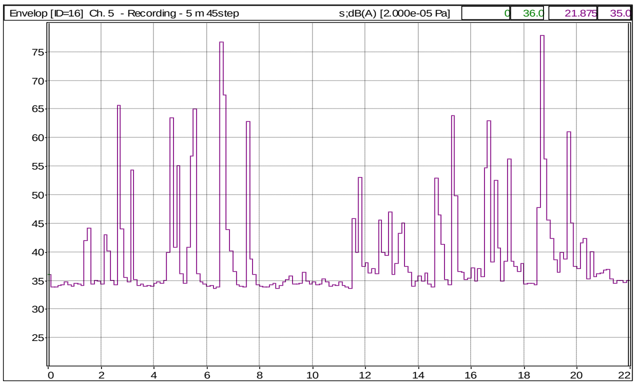

Fig. 13 *The noise mechanism SIG 716 rifles 7.62 mm with a silencer*

The maximum noise level in the work mechanism at each gun with a silencer is about 75 dBA, and every movement of the working parts shall be clearly identified in the recorded sound signal.

3. CONCLUSION

The sound is completely be eliminated only by reducing the pressure of propellant gas at atmospheric conditions, which is practically impossible. The sound is lower for a smaller initial pressure that is less difference between the atmospheric pressure in the barrel.

Simple solutions, such as a large chamber for the expansion of gas or reducing gunpowder impair the basic function of weapons.

When designing silencers theoretical approaches do not coincide with the experimental results. Acoustic theory, explosions and quasi-one-dimensional theory are not entirely applicable to a non-linear bang on the muzzle and the complex dynamics of gas and reflected waves in the silencer chambers.

From all the noise caused by firearms, a silencer works only on the decrease of the shot, or shot at the muzzle.

The overall sound attenuators have the greatest impact at subsonic weapons.

The use of silencers is usually a compromise between noise reduction and accuracy. If a high priority to noise reduction, accuracy and grouping of hits will not meet the strict requirements constructor weapons.

A direct impact on improving the efficiency silencers have reinforcement inside the silencer, increase the number of partitions (size no greater significance), increasing the size of the hole on the tube.

As a secondary sound of a gunshot, sonic boom occurs outside the damper body, to the silencer must be silenced. Neither chamber design will allow the release of these missiles without using subsonic bullet or can not slow down to a speed below the standard speed of sound.

The sound of mechanism at work muffled weapons real pounding in shooting and shutter sound when firing caps, which can be influenced only in the design stage weapons.

Due to the supersonic burst, which is difficult to reduce it, the supersonic missile fired shots through the breakers are very heard. Yet the importance of thugs in locating disabling projectile (for multiple refraction of sound in it).

Modern equipment for determining the location of origin of the shot, using a combination of equipment sensitive microphone IC detectors, special laser and GPS can determine the location from which the firing was carried out in which there is a ballistic shot. When operating this equipment, in addition to ballistic shot, detected by the human eye (and distinguishes him from the animal), the position of the grain in the air (based on temperature difference), optical sighting devices, laser rangefinder work etc.

Therefore, in the real world suppressoion weapons in which there is a ballistic shot because the velocity at the muzzle is always subsonic (about 290 m/s). For them the silencer of the weapons without which a weapon is used.

REFERENCES

[1] S. Jaramez and D. Micković, "Inside Ballistics ", *Faculty of Mechanical Engineering*, Belgrade, 2002.

[2] S. Janković, "Aerodynamics missiles", *Faculty of Mechanical Engineering*, Belgrade, 1979.

[3] J. Marinković, “Inside Ballistics”, P*R Srbija*, 1950.

[4] ISO 17201 Acoustics – Noise from shooting ranges.

[5] ISO 9613 Acoustics – Attenuation of sound during propagation outdoors

[6] M. Vasić, “Special and muffled weapon”, Belgrade 1988.

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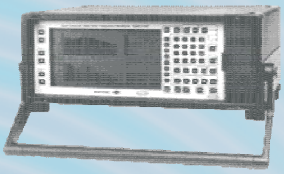
Environmental noise & Occupational noise - measurement and analyis

- Rating noise level;
- Frequency analyis;
- Statistical analysis;



Human vibration

- Hand-arm vibration;
- Whole body vibration;

Room acoustics

- Time reverberation;
- Airborne sound reduction index
- Impact sound reduction index;

Urban noise

- Noise monitoring;
- Noise zoning;
- Strategic noise maps;

Sound power of noise sources

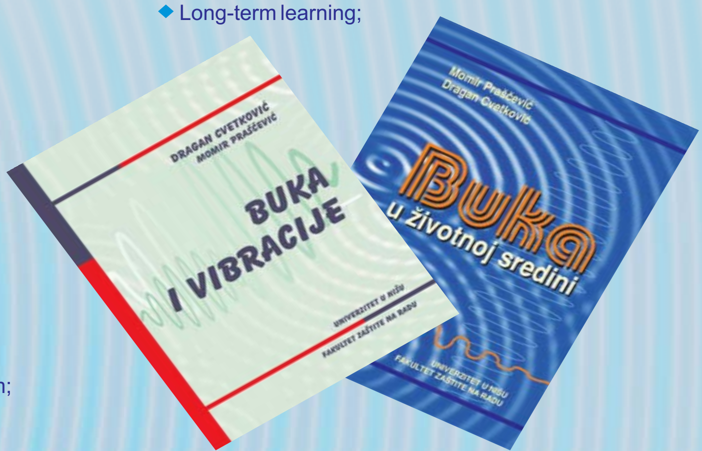
- Sound pressure method;
- Sound intensity method;

Predictive/preventive maintenace of machine

- Vibration condition monitoring;
- Vibrodiagnostics;
- Balancing of rotating machine;

Education

- Workshops;
- Courses;
- Long-term learning;



Design of noise and vibration control systems

- Design of noise insulation and absorption system;
- Design of vibration insulation and absorption system;
- Design of room acoustics;

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Faculty of Mechanical Engineering
25th International Conference
NOISE AND VIBRATION
Tara, 27 - 29. 10. 2016.

LONG-TERM ROAD TRAFFIC NOISE MEASUREMENTS AT THE MAIN STREETS OF NIŠ CITY

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Abstract – *Environmental noise level monitoring in Serbia is performed in several cities and it is pursuant to the Law on Environmental Noise Protection and the accompanying regulations. Although these regulations are in accordance with the national standards, the methodology of noise monitoring varies in different cities. The issues which differ include the following: the number of measurement spots; the number of daily, weekly, and monthly measurement intervals, the duration of measurement intervals, measurement parameters and noise indicators used for noise evaluation. Different measurement procedures are the consequence of diverse city configurations, traffic structure, traffic flow, locations of noise-sensitive objects, as well as diverse contribution of noise sources. The road traffic noise level monitoring in the City of Nis has been organized from 1995 until today based on short-term measurements. The values of noise indicators are calculated based on these short-term measurements. The two newly purchased noise monitoring terminals by the Noise and Vibration Laboratory of the Faculty of Occupational Safety in Nis, enabled the long-term noise measurements. The procedure of permanent and semi-permanent road traffic noise measurements at eight locations in the City of Nis has been carried out since January 1, 2014. The results of long-term road noise measurements at the main streets of Niš city will be presented in this paper.*

1. INTRODUCTION

Noise pollution caused by road traffic represents a major problem in the environment of most urban areas. However, the problem of road traffic noise has not been approached properly so far, and not enough attention has been paid to it in spite of the fact that it has a great impact on the quality of life of the endangered population. Reasons for such an approach could be found in the definition of noise as a subjective experience of various external events, in its specific character, as well as in the difficulties connected to relating the causes with the effects it has on general health.

The latest data related to the environmental noise pollution [1], collected from the first round of strategic noise mapping of the European Union agglomerations, indicate that 54% of the population in urban areas (56,001,200 inhabitants) is exposed to L_{den} noise levels above 55dB, whereas 15% of the population (15,754,500 inhabitants) is exposed to L_{den} noise levels above 65dB. In addition to this, additional 33,437,244 inhabitants outside agglomerations live in areas where L_{den} noise levels exceed 55dB and 7,657,083 live in areas where L_{den} noise levels exceed 65dB. Out of the total of 89,438,444 inhabitants exposed to L_{den} noise levels above 55dB, almost 89 million are exposed to the traffic noise [1].

The conditions related to noise pollution in the city of Niš are in many ways similar to conditions in other urban environments. Collecting information on traffic characteristics and noise levels and updating it over a longer period has proven to be crucial to the evaluation and management of environmental noise. Furthermore, measurement and evaluation of traffic noise are important activities which may result in the development of efficient methods for noise control.

Data on traffic noise levels in the city of Nis have been systematically collected and analysed through the project of monitoring the noise level during a number of years starting from 1995 [2-5]. The road traffic noise level monitoring was based on short-term monitoring.

The obtained results give us an insight into the current condition of the noise level at specific locations, allowing us to compare them to previous measurement results and use this to evaluate tendencies related to possible changes in the future.

Two newly purchased noise monitoring terminals in Noise and Vibration Laboratory enable the long-term noise measurements.

The procedure of permanent and semi-permanent road traffic noise monitoring, starting from January 1, 2014 according to guidelines given in standards SRPS ISO 1996-1 [6] and SRPS ISO 1996-2 [7] and IMAGINE document [8] has been carried out eight locations.

The first results of one-year long-term measuremts were published in some papers [9-11]. Some od the results are publicaly available at web site http://www.znrfak.ni.ac.rs/BVLab-KMB/KMB-Home.html.

The results avialaibel until time of publication of this paper will be presented in the paper.

2. MEASUREMENT LOCATIONS FOR LONG-TERM MEASUREMENTS

The procedure of long-term measuremetas is realized as semi-permanent noise monitoring with different monitoring time. Semi-permanent monitoring, ranging from a three months to two years is more cost-effective monitoring than permanent noise monitoring that includes 24 hours a day, 365 days a year noise measurements using a permanently installed noise monitoring terminal at one location.

Brüel&Kjær's Environmental Noise Management System [9-11] was used for long-term noise monitoring. This system consists of Environmental Noise Management System Software, Type 7843, two Noise Monitoring Terminals (NMT), Type 3639B and one Weather Station, Type WXT520.

Both NMTs are equipped with GPRS router and GPS receiver. One of the terminals (marked as NMT-1) is equipped with weather station, which enable measurement of the following meteorological parameters: temperature, humidity, air pressure, wind velocity, wind direction and rainfall.

The procedure of long-term measurements is realized at eight measurements locations. There are the different methods for the selection measurement locations [12]. The choice of measurement locations was done in accordance with population and residential location, characteristics of land-uses and road functions and structure. The distribution of the measurements locations is given in Fig. 1.

The mark and basic information about measurement locations are given below.

Network of NMT locations is shown in Fig. 1

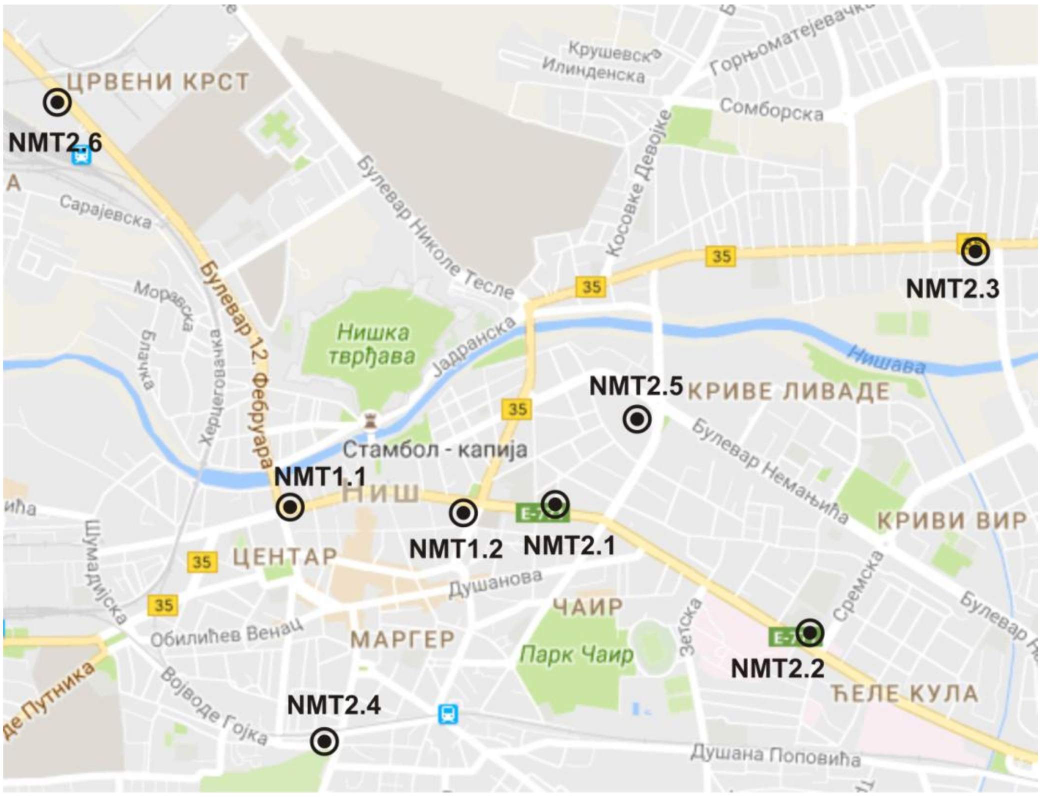

Fig. 1 *Network of NMT locations*

NMT1.1: Intersection of Kneginje Ljubice street and Generala Milojka Lešjanina street

Latitude: 43° 19' 12.8"

Longitude: 21° 53' 27.6"

Altitude: 195.3 m

Microphone height: 4 m

Mounting type: the lighting pole

Monitoring time: two years (01.01.2014 - 31.12.2015)

NMT1.2: Intersection of Kralja Stefana Prvovečanog street and Vožda Karađorđa street

Latitude: 43° 19' 14"

Longitude: 21° 54' 01"

Altitude: 197 m

Microphone height: 4 m

Mounting type: the lighting pole

Monitoring time: in progress (01.01.2016 -)

Plan of monitoring time: one year

NMT2.1: Primary school "Vožd Karađorđe" near Vožda Karađorđa street

Latitude: 43° 19' 13"

Longitude: 21° 54' 13.2"

Altitude: 196.8

Microphone height: 4 m

Mounting type: the lighting pole

Monitoring time: six months (01.01.2014 - 30.06.2014)

NMT2.2: Faculty of Medicine near Dr Zorana Đinđića street

Latitude: 43° 19’ 12”

Longitude: 21° 53’ 27”

Altitude: 197.1

Microphone height: 4 m

Mounting type: the separate pole

Monitoring time: nine months (01.07.2014 - 31.03.2015)

NMT2.3: Residential building near Knjaževačka street

Latitude: 43° 19’ 46”

Longitude: 21° 55’ 58”

Altitude: 212

Microphone height: 4 m

Mounting type: facade

Monitoring time: three months (01.04.2015 - 30.06.2015)

NMT2.4: Commercial building near railway Niš-Sofia-Niš

Latitude: 43° 18’ 46”

Longitude: 21° 53’ 36”

Altitude: 205

Microphone height: 4 m

Mounting type: facade

Monitoring time: six months (01.07.2015 -31.12.2015)

NMT2.5: Day nursery “Bamby” near Bulevar Nemanjića street

Latitude: 43° 19’ 26”

Longitude: 21° 54’ 31”

Altitude: 196

Microphone height: 4 m

Mounting type: the lighting pole

Monitoring time: six months (01.01.2016 -30.06.2016)

NMT2.6: Municipality building near 12. februara street

Latitude: 43° 20’ 2.7”

Longitude: 21° 52’ 51.9”

Altitude: 206

Microphone height: 4 m

Mounting type: the lighting pole

Monitoring time: in progress (01.07.2016 –)

Plan of monitoring time: six months

3. RESULTS OF ROAD TRAFFIC NOISE MONITORING

Monthly values of noise indicators for all locations as well as the results of statistical analysis (energetic mean value, standard deviation and maximum deviation of individual values from the energetic mean value) are shown in following tables.

The main values of noise indicators for all locations are shown in Fig. 2.

Fig. 1 *The values of* $L_{day}/L_{evening}/L_{night}/L_{den}$ *in dB*

Table 1 *The monthly noise indicators in dB for NMT-1.1*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
January 2014	73.1	71.9	67.9	75.9	71.7
February 2014	73.1	71.9	67.7	75.8	71.7
March 2014	73.3	72.1	67.9	76.0	71.9
April 2014	73.4	72.4	68.3	76.3	72.0
May 2014	73.3	72.3	68.1	76.2	71.9
June 2014	73.0	72.0	68.1	76.0	71.7
July 2014	72.8	72.2	67.8	75.8	71.5
August 2014	72.7	71.9	68.2	76.0	71.5
September 2014	73.1	72.0	67.9	75.9	71.7
October 2014	73.2	72.1	68.0	76.0	71.9
November 2014	73.0	72.0	67.6	75.7	71.6
December 2014	73.3	72.4	68.2	76.2	72.0
mean value	**73.1**	**72.1**	**68.0**	**76.0**	**71.8**
σ	0.20	0.17	0.22	0.17	0.18
max. deviation	0.4	0.3	0.4	0.3	0.3

Table 2 *The monthly noise indicators in dB for NMT-1.1*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
January 2015	72.8	71.4	68.5	76.0	71.5
February 2015	72.8	71.8	67.4	75.5	71.4
March 2015	73.2	72.1	67.8	75.9	71.8
April 2015	72.7	71.6	67.3	75.4	71.3
May 2015	72.5	71.4	67.3	75.3	71.2
June 2015	72.5	71.7	68.0	75.8	71.3
July 2015	72.1	71.3	67.5	75.3	70.9
August 2015	72.1	71.3	67.7	75.4	70.9
September 2015	72.6	71.5	67.4	75.4	71.3
October 2015	73.1	72.0	67.8	75.9	71.7
November 2015	72.9	71.6	67.7	75.7	71.5
December 2015	73.0	72.0	69.3	76.7	71.9
mean value	**72.7**	**71.6**	**67.8**	**75.7**	**71.4**
σ	0.35	0.28	0.58	0.41	0.31
max. deviation	0.6	0.4	1.5	1.0	0.5

Table 3 *The monthly noise indicators in dB for NMT-1.2*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
January 2016	69.4	69.3	65.8	73.4	68.5
February 2016	69.2	68.5	64.8	72.6	68.1
March 2016	69.4	69.0	65.0	72.8	68.3
April 2016	69.0	68.7	64.7	72.5	67.9
May 2016	69.2	69.4	65.6	73.2	68.3
June 2016	69.0	69.1	64.9	72.7	68.0
July 2016	68.8	68.7	65.4	72.9	67.9
August 2016	68.8	69.1	67.4	74.2	68.5
September 2016	69.5	70.1	65.5	73.3	68.6
mean value	**69.1**	**69.1**	**65.5**	**73.1**	**68.2**
σ	0.26	0.48	0.82	0.53	0.27
max. deviation	0.4	1.0	1.9	1.1	0.4

Table 4 *The monthly noise indicators in dB for NMT-2.1*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
January 2014	70.3	69.9	67.4	74.7	69.4
February 2014	70.2	69.7	66.7	74.1	69.2
March 2014	70.6	69.8	66.7	74.2	69.5
April 2014	70.5	70.2	67.2	74.6	69.6
May 2014	70.6	70.3	66.8	74.4	69.6
June 2014	70.1	69.7	66.6	74.0	69.1
mean value	**70.4**	**69.9**	**66.9**	**74.3**	**69.4**
σ	0.19	0.26	0.35	0.26	0.18
max. deviation	0.3	0.4	0.5	0.4	0.3

Table 5 *The monthly noise indicators in dB for NMT-2.2*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
July 2014	63.5	63.0	57.6	66.1	62.2
August 2014	62.2	62.0	57.5	65.5	61.1
September 2014	63.1	62.5	57.9	66.0	61.8
October 2014	63.4	62.8	57.9	66.2	62.1
November 2014	63.3	62.8	57.9	66.2	62.1
December 2014	63.7	63.0	58.2	66.5	62.4
January 2015	62.8	61.7	57.4	65.5	61.4
February 2015	63.1	62.7	57.9	66.1	61.8
March 2015	63.6	63.0	58.3	66.5	62.3
mean value	**63.2**	**62.6**	**57.8**	**66.1**	**61.9**
σ	0.46	0.47	0.30	0.36	0.43
max. deviation	1.0	1.0	0.4	0.6	0.8

Table 6 *The monthly noise indicators in dB for NMT-2.3*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
April 2015	62.6	61.9	57.2	65.4	61.2
May 2015	62.0	61.3	56.8	64.9	60.7
June 2015	62.1	61.5	56.9	65.0	60.8
mean value	**62.2**	**61.6**	**57.0**	**65.1**	**60.9**
σ	0.32	0.31	0.21	0.26	0.26
max. deviation	0.4	0.3	0.2	0.3	0.3

Table 7 *The monthly noise indicators in dB for NMT-2.4*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
July 2015	62.9	62.2	60.4	67.4	62.1
August 2015	62.1	61.4	59.8	66.8	61.3
September 2015	63.0	61.8	58.8	66.4	61.8
October 2015	63.7	61.9	59.2	66.8	62.3
November 2015	63.6	61.2	59.8	67.0	62.3
December 2015	63.8	62.4	60.6	67.7	62.7
mean value	**63.2**	**61.8**	**59.8**	**67.0**	**62.1**
σ	0.65	0.46	0.69	0.47	0.48
max. deviation	1.1	0.6	1.0	0.7	0.8

Table 8 *The monthly noise indicators in dB for NMT-2.5*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
January 2016	65.5	64.3	59.8	68.8	64.2
February 2016	65.5	64.8	59.3	68.2	64.4
March 2016	65.7	64.7	59.4	68.0	64.2
April 2016	65.0	64.5	59.4	67.6	63.6
May 2016	65.2	65.1	59.6	68.0	64.0
June 2016	64.6	64.9	59.4	67.8	63.5
mean value	**65.3**	**64.7**	**59.5**	**68.1**	**64.0**
σ	0.40	0.29	0.18	0.41	0.36
max. deviation	0.7	0.4	0.3	0.7	0.5

Table 9 *The monthly noise indicators in dB for NMT-2.6*

	L_d	L_e	L_n	L_{den}	$L_{eq,total}$
July 2016	67.9	67.5	63.9	71.5	66.9
August 2016	68.2	67.9	64.1	71.8	67.1
September 2016	68.7	69.6	64.0	72.3	67.8
mean value	**68.3**	**68.3**	**64.0**	**71.9**	**67.3**
σ	0.40	1.12	0.10	0.40	0.47
max. deviation	0.4	1.3	0.1	0.4	0.5

The monthly values of noise indicators for all locations are slightly different from the energetic mean values of noise indicators for observation interval except for the case of occasional occurrences such as New Year`s celebration (NMT1.1. – December 2015), celebration of Olympic champion in water polo (NMT1.2 – August 2016).

4. IN PLACE OF CONCLUSION – NEW APPROACH FOR ENVIRONMENTAL NOISE ASSESSMENT

It currently remains difficult for people to understand the environmental noise data due to various noise indicators that are expressed in decibel unit which is logarithmic in nature, and usually complicated to explain and relatively far-removed from perception of people. Also, the noise indicators very often are expressed in dB(A), which further complicates the understanding of noise indicators values.

Two French organizations specialized for management and organization of urban noise observatories in France, have worked on a proposal for a new index closer to the perception of the people [13, 14]. They suggested a new environmental noise index called Harmonica (HARMOnised Noise Information for Citizens and Authorities) index. The Harmonica index is based on measurement data obtained by noise monitoring and take into account both the overall environmental noise load and noise peaks from sudden noise events.

The Harmonica index is calculated based on one-hour time sample of A-weighted, equivalent continuous sound level sampled with 1 second interval, and it takes into account the two major components that affect the sound environment.

The Harmonica index is an adimensional index based on a scale of 0 to 10. The Harmonica index is graphically represented as a triangle (BGN component) on top of a rectangle (EVT component). Three colors (green, orange and red) are used for color representation of the Harmonica index. The color scale is shown in Table 10.

Table 10. *The color scale for Harmonica index*

Color	Day (from 6 am to 10 pm)	Night (from 10 pm to 6 am)	Harmonica index score
green	between 0 and 4	between 0 and 3	Quiet
orange	between 4 and 8	between 3 and 7	Noisy
red	over 8	over 7	Very noisy

The detailed information about Harmonica index, calculation procedure and the results of environmental noise assessment by Harmonica index in the city of Nis are given in papers [15, 16].

Two example of Harmonica index calculation are given in Fig. 3 and Fig.4.

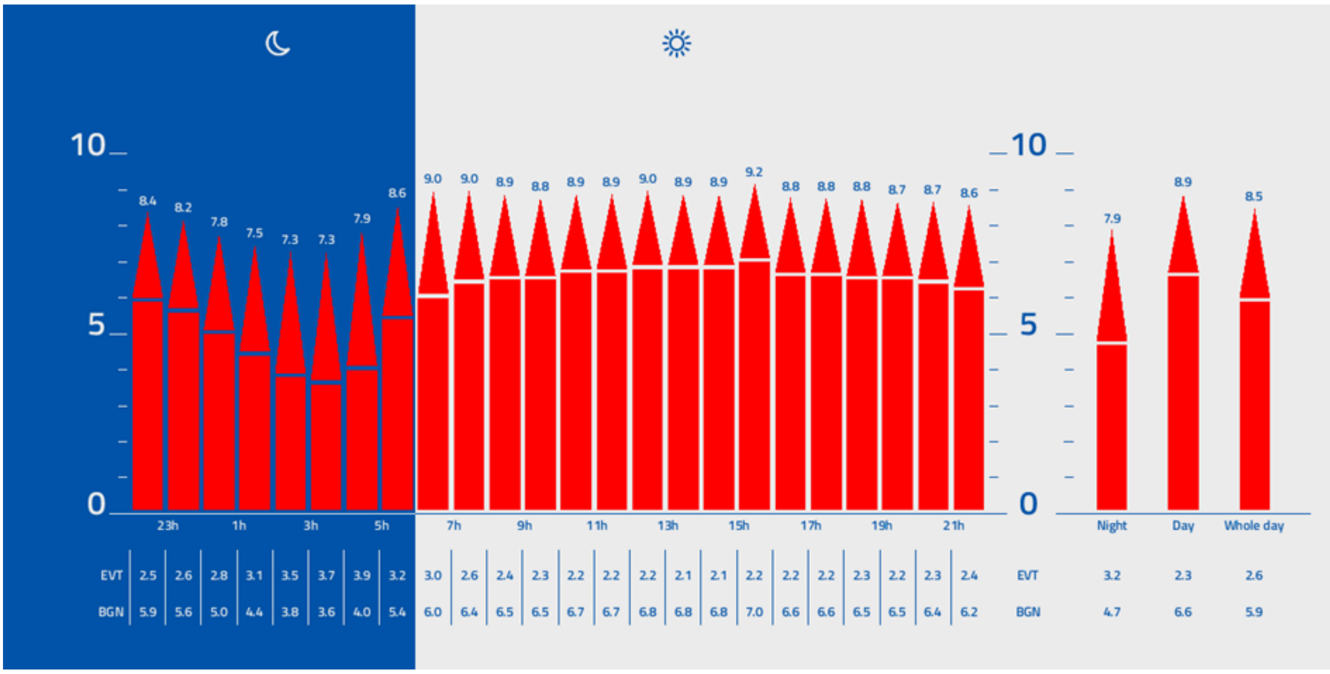

Fig. 3. *The averaged hourly values of Harmonica indices for NMT-1.1 (April 20, 2015- April 26, 2015)*

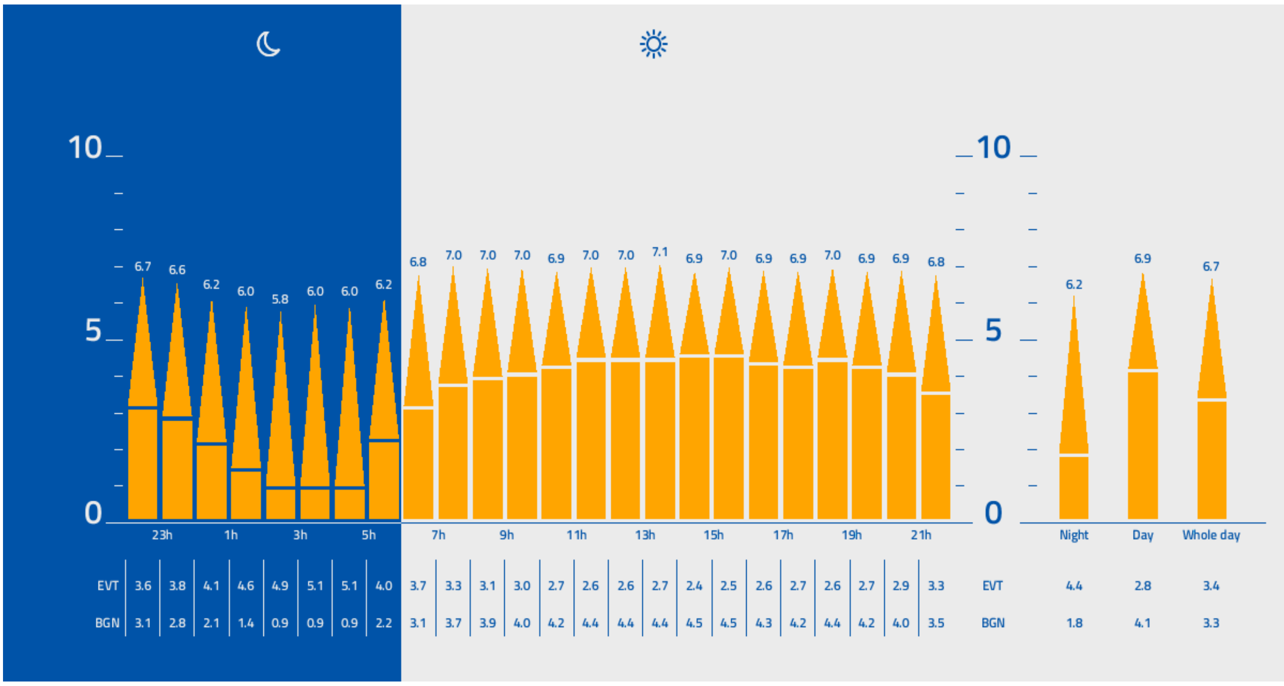

Fig. 4. *The averaged hourly values of Harmonica indices for NMT-2.3 (April 20, 2015- April 26, 2015)*

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REFERENCES

[1] P. Vos, G. Licitra, Noise maps in the European Union: An overview. In G. Licitra (Ed.), Noise mapping in EU: Models and Procedures, 1st edn., CRC Press, USA, 2012.

[2] M. Prascević, "Systematic Environmental Noise Measurement of Nis", *Proc. of XV Yugoslav and III International Conference "Noise and Vibration in Living and Working Environment"*, Niš, Serbia, pp. 35-38, 1995

[3] D. Cvetković, A. Deljanin, M. Prascevic, "Community noise levels survey of Nis", *Proc. of the International Congress on Noise Control Engineering "INTER NOISE"*, Budampest, Hungary, pp. 815-818. 1997

[4] D. Mihajlov, M. Prascevic, D. Cvetkovic, "Results of the monitoring and assessment of the state of noise level in Nis in 2007", *Proc. of 21st Conference with International Participation "Noise and Vibration"*, Nis, Serbia, ID: 21-23, 2008

[5] D. Mihajlov, M. Prascevic, D. Cvetkovic, "An analysis of the environmental noise levels on the territory of the city of Nis", Proc. of 23rd National and 4th International Conference "Noise and Vibration", Nis, Serbia, pp. 49–58, 2012

[6] SRPS ISO 1996-1: 2010, Acoustics - Description, measurement and assessment of environmental noise - Part 1: Basic quantities and assessment procedures.

[7] SRPS ISO 1996-2: 2010, Acoustics - Description, measurement and assessment of environmental noise -Part 2: Determination of environmental noise levels.

[8] IMAGINE Project, Deliverable 5, "Determination of L_{den} and L_{night} using measurements", 2011. [Online]. Available: http://www.certificacioacustica.cat/Documents/Articles/D5_IMA32TR-040510-SP08.pdf

[9] M. Praščević, D. Mihajlov, D. Cvetković, „Permanent and semi-permanent noise monitoring – first results in the city of Nis", in Proc. of 24th International Conference „Noise and Vibration", ISBN: 978-86-6093-062-2, pp. 33-40, Niš, 2014

[10] M. Praščević, D. Mihajlov, „Noise indicators determination based on long-term measurements", Facta universitatis, Series: Working and Living Environmental Protection, Vol. 11, No. 1, pp. 1-11, 2014

[11] D. Mihajlov, M. Praščević, „Permanent and Semi-permanent Road Traffic Noise Monitoring in the City of Nis (Serbia)", Journal of low frequency noise, vibration and active control, Vol. 34, No. 3, ISSN: 0263-0923, University of Sheffield, pp. 251-268, http://dx.doi.org/10.1260/0263-0923.34.3.251, 2015

[12] H. Doygum, D. K. Gurun, Analysing and mapping spatial and temporal dynamics of urban traffic noise pollution: a case study in Kahramanmaraş, Turkey. Environmental Monitoring and Assessment, 142, 65-72. 2008

[13] C. Mietlicki, F. Mietlicki, C. Riberio, P. Gaudibert, "The HARMONICA project, new tools to assess environmental noise and better inform to the public", *Proceedings of Forum Acousticum*, Krakow, Poland, 2014

[14] F.Mietlicki, P. Gaudibert, "The HARMONICA project (HARMOnized In-formation for Citizens and Authorities)", *Proceedings of Inter-noise*, New York, USA, 2012

[15] M. Praščević, D. Mihajlov, D. Cvetković, „Assessment of environmental noise by Harmonica index – case study: the city of Niš", Applied Mechanics and Materials, Vol. 801, Trans tech publications Inc., ISBN 13: 978-3-03835-628-8, pp 51-59, http://dx.doi.org/10.4028/www.scientific.net/AMM.801.51, 2015

[16] M. Praščević, D. Mihajlov, D. Cvetković, „The Correlation between Harmonica Indices and Noise Indicators", ANALELE UNIVERSITĂȚII "EFTIMIE MURGU" REȘIȚA ANUL XXII, NR. 2, ISSN 1453 – 7397, University of Resita, pp 306-317, http://anale-ing.uem.ro/2015/230.pdf, 2015

ENVIRONMENTAL NOISE IN CITY STREETS AND CAR-FREE ZONES IN NOVI SAD

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Abstract *- Objectives: Using SRPS ISO 1996-1:2010 / SRPS ISO 1996-2:2010, IPHV performed environmental noise monitoring in Novi Sad, using monitoring spots due to urban space purpose and Urban Plan, during June 2015 - May 2016. Materials and methods: IPHV performed 36 24-hour measurements in "city center and city street zones" on three monitoring spots. Two of them were road traffic spots - area in front of the Government building (Bulevar Mihajla Pupina) and settlement of Telep (Vršačka ulica). Third spot was car-free zone in City center (Zmaj Jovina ulica). Results: In car-free zone, mean equivalent noise level (L_{Aeq}) is statistically significant lower in relation to street zones L_{Aeq} in Bulevar Mihajla Pupina (p=0,002) and Telep (p=0,000). In car-free zone L_{Aeq} at 630Hz is statistically significant* higher *(p=0,001), while L_{Aeq} at 1000Hz is statistically significant lower (p=0,000) in relation to L_{Aeq} street zones. Conclusion: Except for the measured noise values, car-free zone noise is also different reffered to street zones noise, according to frequency analysis. There is need to consider national regulation that in same manner treats the noise level in city center and car-free zones. Key words: Noise, health, monitoring, indicator*

1. INTRODUCTION

Noise in the environment can cause adverse effect to human health [1, 2, 3]. The effects of environmental noise could be effects on sleep, annoyance, cognitive impairment, cardiovascular disease, hearing impairment, tinnitus, adverse birth outcomes and mental health and wellbeing [4]. World Health Organization (WHO) indicates that more than one million healthy years of life are lost every year from road traffic-related noise in the west Europe [5]. Also, noise pollution is a major environmental health problem in Europe, road traffic is the most dominant source of environmental noise with an estimated 125 million people affected by noise levels greater than 55 decibels (dB) L_{den} (6).

The aim of environmental noise monitoring is human health protection and also development and improvement of environment. Monitoring was implemented according to the current legal and professional basis in order to determine status of the environment, for the selection of preventive actions to protect and improve human health and environment, based on the signed by the Environmental Protection city Council of Novi Sad and the Institute of Public Health of Vojvodina.

2. MATERIALS AND METHODS

The national legal basis defines the allowed environmental noise level, noise measurement methods, conditions for noise measurement organizations and content of noise sources documents.

IPHV as authorized and accredited institution (SRPS ISO/IEC 17025:2006, SRPS ISO 9001:2008, SRPS ISO 14001:2006) performed environmental noise monitoring in the city of Novi Sad according to standardized methodology [7, 8] and due to legal and proffesional basis [9, 10, 11, 12, 13, 14].

During June 2015 - May 2016 IPHV performed total 91 24-hour measurements in Novi Sad on 7 monitoring spots, in agreement with city authorities.

In this paper, it is presented data of 36 24-hour measurements in "City Centre, craft, trade and administrative areas with apartments, roads, highways and city street zones" (city center and city street zones) on 3 monitoring spots. Two of them were road traffic spots - area in front of the Government building (Bulevar Mihajla Pupina) and settlement of Telep (Vršačka ulica) (Figures 1 and 2). Third spot was car-free zone in City center (Zmaj Jovina ulica) (Table 1, Figure 3).

Table 1 *Monitoring spots in city center and city street zones*

No	Spot name	Coordinates	Purpose
1	Area in front of the Government building (Bulevar Mihajla Pupina)	N 45°15’09,81“; EO 19°51’03,38“	city street zone
2	Settlement of Telep (Vršačka ulica)	N 45°15’10,41“; EO 19°52’29,86“	city street zone
3	City Center (Zmaj Jovina ulica)	N 45°15’21,55“; EO 19°50’48,18“	car-free zone

Statistical analysis is performed using SPSS ver.21.0.

Fig. 1 *Photography, area in front of the Government building (Bulevar Mihajla Pupina), city street zone*

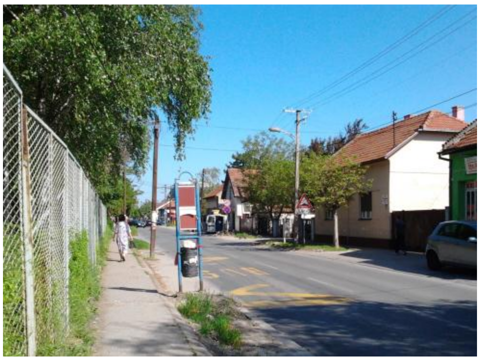

Fig. 2 *Photography, settlement of Telep (Vršačka ulica), city street zone*

Fig. 3 *Photography, Zmaj Jovina ulica, car-free zone*

In this paper, it is considered equivalent noise level (L_{Aeq}), compared car-free zone with city street zones.

The measuring system used in environmental noise monitoring is: Brüel&Kjær type 2250; Brüel&Kjær type 3535-A (all weather suitcase); Brüel&Kjær Outdoor Microphone type 4952; BZ 5503 Utility Software; Noise Explorer Type 7815; KIMO AMI 300 (for microclimate measurements).

Statistical data analysis were performed using SPSS ver.21.0.

3. RESULTS

3.1 Data for city street zones

Table 2a *Data for city street zone - Basic noise indicator - Area in front of the Government building (Bulevar Mihajla Pupina) during June 2015 - May 2016*

Date	L_{day} dB	$L_{evening}$ dB	L_{night} dB	L_{den} dB
27.06.2015	66.2	64.4	61.6	69.2
11.07.2015	64.0	66.1	64.7	71.2
08.08.2015	62.9	63.5	60.9	68.0
13.09.2015	64.4	63.8	62.4	69.3
31.10.2015	64.7	63.4	60.9	68.3
28.11.2015	66.8	65.3	63.6	70.7
19.12.2015	65.3	63.7	61.2	68.6
23.01.2016	64.4	63.5	61.0	68.3
13.02.2016	65.0	63.2	60.6	68.1
12.03.2016	65.6	65.0	61.0	68.9
09.04.2016	64.7	63.3	60.9	68.2
14.05.2016	64.6	63.6	61.2	68.5
N of 24h noise measurements	12	12	12	12
Minimum	62.9	63.2	60.6	68.0
Maximum	66.8	66.1	64.7	71.2
Average	64.9	64.1	61.7	68.9
Noise levels exceed limit values	**2 (17%)**	**1 (8%)**	**12 (100%)**	-
Noise levels in accordance with limit values	**10 (83%)**	**11 (92%)**	**0 (0%)**	-

N - nummber

Table 2b *Data for city street zone - L_{Aeq} - Area in front of the Government building (Bulevar Mihajla Pupina) during June 2015 - May 2016*

Date of 24h measurement	L_{Aeq} on frequency of 630 Hz	L_{Aeq} on frequency of 1000 Hz	L_{Aeq} dB
27.06.2015	54.28	56.27	64.82
11.07.2015	54.69	56.20	64.64
08.08.2015	51.78	54.27	62.43
13.09.2015	52.82	55.45	63.75
31.10.2015	52.61	55.45	63.54
28.11.2015	53.08	56.64	65.70
19.12.2015	52.60	55.96	64.00
23.01.2016	52.33	55.93	63.34
13.02.2016	52.15	55.56	63.61
12.03.2016	52.47	55.94	64.39
09.04.2016	52.56	55.71	63.49
14.05.2016	52.71	55.66	63.57
Average (Mean)	52.84	55.75	63.94

Table 3a *Data for city street zone - Basic noise indicator - Settlement of Telep (Vršačka ulica) during June 2015 - May 2016*

Date	L_{day} dB	$L_{evening}$ dB	L_{night} dB	L_{den} dB
25.6.2015	67.4	64.0	60.2	68.8
14.7.2015	65.4	63.3	58.4	67.1
4.8.2015	65.6	63.1	58.1	67.0
8.9.2015	66.9	64.4	59.7	68.4
20.10.2015	66.9	64.7	60.3	68.8
3.11.2015	65.8	64.0	59.4	67.9
10.12.2015	66.6	64.4	61.3	69.1
5.1.2016	68.5	64.7	60.3	69.4
3.2.2016	66.3	65.3	60.6	68.9
2.3.2016	64.7	63.0	59.0	67.1
14.4.2016	66.8	64.6	57.2	67.5
19.5.2016	65.8	63.9	59.2	67.7
N of 24h noise measurements	12	12	12	12
Minimum	64.7	63.0	57.2	67.0
Maximum	68.5	65.3	61.3	69.4
Average	**66.4**	**64.1**	**59.5**	**68.1**
Noise levels exceed limit values	**9 (75%)**	**0 (0%)**	**12 (100%)**	-
Noise levels in accordance with limit values	**3 (25%)**	**12 (100%)**	**0 (0%)**	-

Table 3b *Data for city street zone - Settlement of Telep (Vršačka ulica) during June 2015 - May 2016*

Date of 24h measurement	L_{Aeq} on frequency of 630 Hz	L_{Aeq} on frequency of 1000 Hz	L_{Aeq} dB(A)
25.06.2015	53.19	56.90	65.44
14.07.2015	51.83	55.54	63.66
04.08.2015	51.11	55.09	63.75
08.09.2015	52.64	57.10	65.08
20.10.2015	51.85	56.98	65.16
03.11.2015	50.54	56.23	64.16
10.12.2015	50.99	56.90	65.03
05.01.2016	50.68	56.54	66.39
03.02.2016	52.02	56.50	64.89
02.03.2016	51.56	54.32	63.15
14.04.2016	50.07	54.97	64.82
19.05.2016	53.17	57.04	64.09
Average (Mean)	**51.63**	**56.17**	**64.63**

3.2 Data for car-free zone

Table 4a *Data for car-free zone - Basic noise indicators - City Center (Zmaj Jovina ulica) during June 2015 - May 2016*

Date	L_{day} dB	$L_{evening}$ dB	L_{night} dB	L_{den} dB
21.06.2015	61.7	65.5	63.5	69.8
05.07.2015	57.1	65.9	62.3	69.0
30.08.2015	57.9	66.2	61.8	68.7
27.09.2015	59.1	58.3	61.5	67.4
25.10.2015	64.1	59.7	61.4	68.1
15.11.2015	62.6	58.2	60.0	66.6
06.12.2015	58.8	58.6	61.0	66.9
17.01.2016	56.3	55.2	60.3	66.0
21.02.2016	61.5	56.6	63.6	69.3
20.03.2016	65.8	60.2	65.0	71.1
17.04.2016	62.4	66.3	62.8	69.8
22.05.2016	62.6	68.4	60.7	69.3
N of 24h noise measurements	12	12	12	12
Minimum	56.3	55.2	60.0	66.0
Maximum	65.8	68.4	65.0	71.1
Average	**60.8**	**61.6**	**62.0**	**68.5**
Noise levels exceed limit values	**1 (8%)**	**4 (33%)**	**12 (100%)**	-
Noise levels in accordance with limit values	**11 (92%)**	**8 (67%)**	**0 (0%)**	-

Table 4b *Data for car-free zone - City Center (Zmaj Jovina ulica) during June 2015 - May 2016*

Date of 24h measurement	L_{Aeq} on frequency of 630 Hz	L_{Aeq} on frequency of 1000 Hz	L_{Aeq} dB(A)
21.06.2015	55.65	52.03	63.16
05.07.2015	54.22	51.42	61.68
30.08.2015	54.40	51.43	61.75
27.09.2015	52.37	49.50	59.97
25.10.2015	54.98	52.48	62.82
15.11.2015	52.84	51.10	61.31
06.12.2015	51.37	49.67	59.60
17.01.2016	50.36	48.39	57.95
21.02.2016	53.36	51.31	61.90
20.03.2016	58.06	55.31	64.94
17.04.2016	56.18	52.39	63.44
22.05.2016	56.33	52.84	63.87
Average (Mean)	**54.17**	**51.48**	**61.86**

3.3 Differences between car-free zone and city street zone

In car-free zone, mean equivalent noise level (L_{Aeq}) is statistically significant lower in relation to street zones L_{Aeq} in Bulevar Mihajla Pupina (p=0,002) and Telep (p=0,000) (Table 5, Figure 4)

Table 5 *Equivalent noise level (A) in relation to street zones*

Spot	N	Mean	Std. dev.	Lower bound §	Upper bound §	Min	Max
CFZ*	12	61.86	1.97	60.61	63.12	57.95	64.94
CS**	12	63.94	0.84	63.40	64.47	62.43	65.70
CS***	12	64.63	0.90	64.06	65.20	63.15	66.39
Total	36	63.48	1.76	62.88	64.07	57.95	66.39

§ - 95% confidence interval for mean
CFZ* - car-free zone, City Center (Zmaj Jovina ulica)
CS** - city streets zone, Area in front of the Government building (Bulevar Mihajla Pupina)
CS*** - city streets zone, Settlement of Telep (Vršačka ulica)

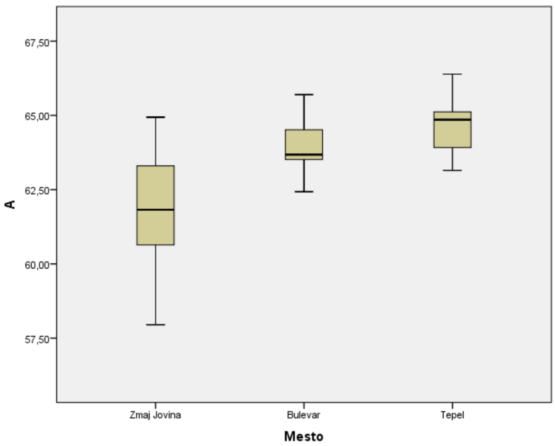

Fig. 4 *Bonferroni post hoc, L_{Aeq}*

In car-free zone L_{Aeq} at 630Hz is statistically significant higher (p=0,001), while L_{Aeq} at 1000Hz *is* statistically significant lower (p=0,000) in relation to L_{Aeq} street zones (Table 6-8, Figure 5-7).

Table 6 *L_{Aeq} at 630Hz in relation to L_{Aeq} at 1000Hz*

Spot	N	Mean	Std. dev.	Lower bound §	Upper bound §	Min	Max
CFZ*	12	54.17	2.23	52.75	55.59	50.36	58.06
CS**	12	55.75	0.58	55.38	56.12	54.27	56.64
CS***	12	56.17	0.95	55.56	56.78	54.32	57.10
Total	36	55.36	1.64	54.81	55.92	50.36	58.06

§ - 95% confidence interval for mean
CFZ* - car-free zone, City Center (Zmaj Jovina ulica)
CS** - city streets zone, Area in front of the Government building (Bulevar Mihajla Pupina)
CS*** - city streets zone, Settlement of Telep (Vršačka ulica)

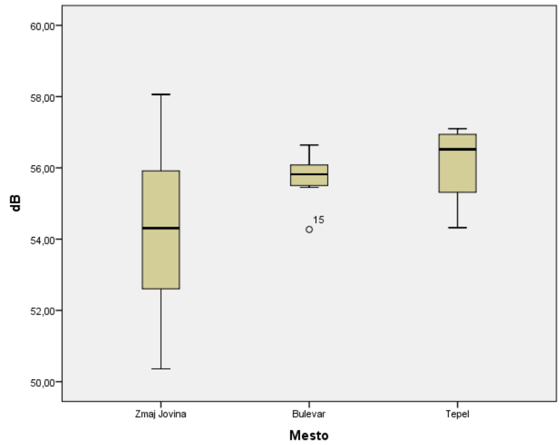

Fig. 5 *Bonferroni post hoc, L_{Aeq} at 630Hz in relation to L_{Aeq} at 1000Hz*

Table 7 *L_{Aeq} at 630Hz in relation to street zones*

Spot	N	Mean	Std. dev.	Lower bound §	Upper bound §	Min	Max
CFZ*	12	54.17	2.23	52.75	55.59	50.36	58.06
CS**	12	52.84	0.83	52.30	53.37	51.78	54.69
CS***	12	51.63	1.01	50.99	52.28	50.07	53.19
Total	36	52.88	1.79	52.27	53.49	50.07	58.06

§ - 95% confidence interval for mean
CFZ* - car-free zone, City Center (Zmaj Jovina ulica)
CS** - city streets zone, Area in front of the Government building (Bulevar Mihajla Pupina)
CS*** - city streets zone, Settlement of Telep (Vršačka ulica)

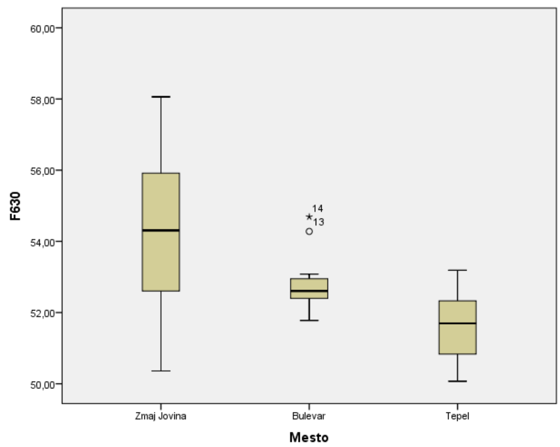

Fig. 6 *Bonferroni post hoc, L_{Aeq} at 630Hz in relation to street zones*

Table 8 *L_{Aeq} at 1000 Hz in relation to street zones*

Spot	N	Mean	Std. dev.	Lower bound §	Upper bound §	Min	Max
CFZ*	12	51.48	1.79	50.34	52.63	48.39	55.31
CS**	12	55.75	0.586	55.38	56.12	54.27	56.64
CS***	12	56.17	0.95	55.56	56.78	54.32	57.10
Total	36	54.47	2.45	53.64	55.30	48.39	57.10

§ - 95% confidence interval for mean
CFZ* - car-free zone, City Center (Zmaj Jovina ulica)
CS** - city streets zone, Area in front of the Government building (Bulevar Mihajla Pupina)
CS*** - city streets zone, Settlement of Telep (Vršačka ulica)

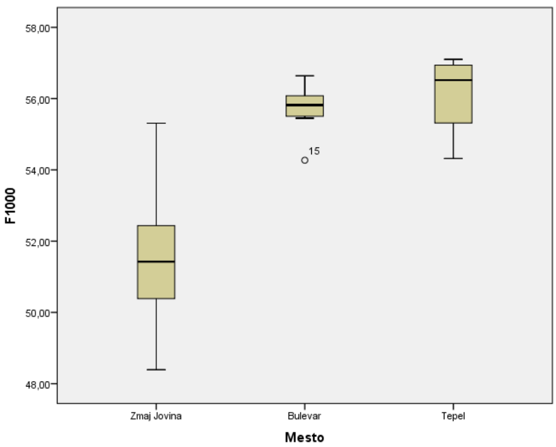

Fig. 7 *Bonferroni post hoc, L_{Aeq} at 1000 Hz in relation to street zones*

4. CONCLUSION

According to frequency analysis, environmental noise in car-free zone noise is different reffered to street zones noise.

There is need to consider national regulation that in same manner treats the environmental noise level in city center and car-free zones.

REFERENCES

[1] The European Parliament and The Council, "Directive 2002/49/EC" (Official Journal of the European Communities, 2002).

[2] E. Zivadinovic, S. Bijelovic, N. Dragic, M. Popovic, S. Milosevic, Z.Lalovic, "Highly Annoyed People With Noise in The city of Novi Sad" (14th DKMT Euroregional Conference on Environment and Health, Abstract of papers and posters in electronic form) (M34) (Szeged, Hungary, May 18-19, 2012).

[3] E. Zivadinovic, S. Bijelovic, N. Dragic, M. Popovic, S. Milosevic, Z. Lalovic, "Environmental Noise As The Hazard In The Environment of The City Of Novi Sad", (23rd National Conference & 4th International Conference Noise And Vibration) (Niš, Serbia, October 17-19, 2012).

[4] WHO Environmental noise guidelines for the European Region. Copyright©(2015) by EAA-NAG-ABAV, ISSN 2226-5147. Available on http://www.na-paw.org/WHO-noise-guidelines-Euronoise-2015.pdf

[5] World Health Organization, Regional Office for Europe and JRC European Commission, "Burden of disease from environmental noise". Quantification of healthy life years lost in Europe. (WHO, Geneva, 2011). Available on http://www.euro.who.int/__data/assets/pdf_file/0008/136466/e94888.pdf

[6] Noise in Europe 2014. © European Environment Agency, 2014 (EEA). ISBN 978-92-9213-505-8. ISSN 1977-8449

[7] SRPS ISO 1996-1:2010, "Acoustics – Description, measurement and estimation of noise in environment – Part 1: Basic units and estimation methods" (Institute for Standardization of Serbia, Belgrade, 2010).

[8] SRPS ISO 1996-2:2010, "Acoustics – Description, measurement and estimation of noise in environment – Part 2: Determination of the noise level in environment" (Institute for Standardization of Serbia, Belgrade, 2010).

[9] National Assemby of the Republic of Serbia, "Law on environmental care" (Official Gazette of RS No 135/04 and 36/09, 2009).

[10] National Assemby of the Republic of Serbia, "Law on environmental noise protection" (Official Gazette of RS", no. 36/2009 and 88/10, 2010).

[11] National Assemby of the Republic of Serbia, "Law on public health" (Official Gazette of RS RS No 72/09, 2009).

[12] Serbian Government, "Regulation on noise indicators, limit values, assessment methods for indicators of noise, disturbance and harmful effects of noise in the environment" (Official Gazette of RS, no. 75/2010, 2010).

[13] Republic of Serbia, Ministry of Environmental Protection and Sppatial Planing, "Rulebook on the methodology for action plans development" (Official Gazette of RS, no. 72/2010, 2010).

[14] Republic of Serbia, Ministry of Environmental Protection and Sppatial Planing, "Rulebook on noise measurement methods, contents and scope of noise measurement reports" (Official Gazette of RS, no. 72/2010, 2010).

Development of methodologies and means for noise protection of urban areas

UrbaNoise

Goals

Provide technical support to implementation of national strategy for protection from environmental noise by:

- Developing certified facility for characterization of noise sources
- Developing software support for local noise mapping and modeling
- Design of modular noise protection means

Project concept

WP-5 Management
WP-4 Promotion

WP-1	WP-2	WP-3
Development of methods for estimation of exposition to noise	Development of materials and means for reduction of noise emission	Design of means for noise protection

Objectives

- Study on dominant noise sources in urban areas
- Design of the database for description of urban noise sources
- Construction of reverberation chamber and semi-anechoic chamber
- Development of software modules for noise modeling
- Study on state–of–the–art of noise protection means
- Design of modular barriers for noise protection

Financial resources

220,000 EUR

- Personnel costs 140,000 EUR
- Equipment 60,000 EUR
- Other 20,000 EUR

Sources:

- Ministry of Education and Science
- City of Niš
- City of Kraljevo

Supported by

Government of the Republic of Serbia
Ministry of Education, Science and Technological Development

Implemented by

Faculty of Mechanical and Civil Engineering
University of Kragujevac

Faculty of Occupational Safety
University of Niš

Faculty of Traffic Engineering
University of Belgrade

University of Nis
Faculty of Occupational Safety
„Politechnica“ University of Timisoara
Faculty of Mechanical Engineering
25th International Conference
NOISE AND VIBRATION
Tara, 27 - 29. 10. 2016.

INFLUENCE OF AIRCRAFT NOISE ON QUIET AREAS

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Abstract – *One of the major environmental issues that have adverse effects on human health and the environment is certainly noise. Different regulations have been adopted with the aim to reduce noise pollution but also to preserve areas that are currently unaffected. Areas not yet disturbed by noise are called quiet areas and their protection should be equally important as noise level reduction. Quiet areas are usually far away from major rail or road traffic corridors. However, moving source such as aircraft could easily increase noise levels in quiet areas if they are under flight trajectories and in the vicinity of the airport. In order to analyse influence of aircraft noise on quiet areas one location in Belgrade was selected as a case study. Several short-term measurements were conducted and results were paired with air traffic data. Difference in noise levels in cases with and without the presence of aircraft noise at one site intended for relaxation of people has been shown and the recommendations how to preserve quiet areas have been given.*

1. INTRODUCTION

Noise is considered to be one of the main environmental issues in the modern society. Transport and industry activities emit noise to which people are inevitably exposed everyday. Long term exposure to significant noise levels can damage human health and negatively affect environment. Different regulations at national and international level have been adopted in order to reduce noise pollution as well as to preserve areas that are currently unaffected. Protecting areas not yet disturbed by noise can bring significant environmental and health benefits [1]–[3]. Environments with good acoustic quality, called quiet areas, may be found, not only in countryside, but also inside busiest cities in the world.

Environmental Noise Directive [4] defines quiet area in an agglomeration as an area, delimited by the competent authority, which is not exposed to a value of Lden or of another appropriate noise indicator greater than a certain value set by the Member State, from any noise source. Similarly, quiet area in open country is defined as an area, delimited by the competent authority, that is undisturbed by noise from traffic, industry or recreational activities. Obligation of every EU Member State is to identify and preserve quiet areas in order to protect environmental health and well-being.

Perception of the landscape, including the distance to the noise sources and the degree of naturalness of the landscape, has been considered crucial to identifying potential quiet areas [1]. When considering the distance to the noise sources, it should be taken into account the specific aspects of each transportation mode. Even thought parks, forests or lakes that are not in immediate proximity of an airport (or some other modes of transport or industrial facilities) should be identified as quiet areas, they could be significantly affected by aircraft overflights which could increase the noise levels over such territories. In this case, the distance to the noise source depends on flight trajectories but also on terrain, including the airport elevation and altitude of the noise receiver. Such characteristic of aircraft noise indicates that it could significantly influence the quality of soundscape at some areas for leisure and recreation.

Having that in mind, the aim of this study was to investigate the influence that aircraft noise has on quiet areas. The goal was to show the difference in noise levels in cases with and without the presence of aircraft noise at one site intended for relaxation of people. For that purpose, the artificial Lake Sava (called Ada Ciganlija) in Belgrade was used as a case study since it has characteristics of a quiet area with frequent aircraft overflights. Short-term measurements were conducted for 18 aircraft overflights during measurement time and for the periods without the presence of aircraft noise.

This paper is organized as follows. Section 2 describes methodology, particularly emphasizing quiet area selection process, measurement location, equipment used, air traffic data, and meteorological situation. By analysing the measured values combined with the air traffic data, Section 3 provides the discussion of results obtained. Section 4 contains conclusions.

2. METHODOLOGY

In order to investigate the influence that aircraft noise has on quiet areas, several steps have been performed. Firstly, quiet areas in Belgrade have been identified and the most suitable for this study has been selected. Secondly, the precise measurement location has been chosen and the measurement has been performed. Subsequently, air traffic data have been collected and paired with measured data for every flight. The description of every step is briefly given below.

2.1 Quiet area selection

The first step in this research was to find the suitable area that could be qualified as „quiet“. Since Directive 2002/49/EC has been only partially transposed into Serbian legislation by the

Law on Environmental Noise Protection (Official Gazette of the Republic of Serbia no 36/2009, 88/2010) and secondary legislation, quiet areas have not yet been determined for agglomerations in Serbia [5]. Relevant guidelines and current practices indicate that approaches, methods and indicators for the identification of quiet areas differ among countries [1], [6]. Various parameters such as different acoustic indicators, functional use of area, distance from motorway and agglomeration, soundscape and size of an area as well as visual aspect of surrounding could be used as selection criteria for quiet areas. Indicators and range criteria for proposed selection criteria type are shown in Table 1.

Table 1 *Selection criteria for quiet areas (not-limitative set)* [6]

Type	Indicator	Range criteria Urban (dB)	Range criteria Open country (dB)
Acoustic indicators	Leq,24h	40	25–45
	Lden	50–55	-
	L50	-	35–45
	L90	-	30
	L95	30	-
	Lday	45–55	30–40
Functional	Recreation	Moderate intensive activity	Passive activity
	Nature protection	Moderate	Priority
	Health protection/restoration	Health protection	Restoration priority
Distance	From motorway	-	4–15 km
	From agglomeration	-	1–4 km
Soundscape	Perceived acoustic quality/appreciation	-	-
Size	-	100–100 000 m2	0.1–100 km2
Visual	Areas with established values in official documents, e.g. land use plans or nature conservation plans	-	-

Several locations in Belgrade that could meet such requirements have been identified and they are: Košutnjak, Avala, artificial Lake Sava, Kalemegdan Park, Great War Island, Zvezdara forest, etc. From the above mentioned areas, artificial Lake Sava has been chosen since it seemed as the most suitable location to study the influence of aircraft noise due to the large daily number of visitors especially during summer season and frequent overflights.

2.2 Measurement location

The artificial Lake Sava is located around 9km from the Belgrade Nikola Tesla airport, measured from the start of the take-off roll from runway 12 in the direction of take-off (Fig. 1). Surface elevation of the Lake Sava is 78m, while elevation of Belgrade Nikola Tesla airport is 102m.

Fig. 1 *Distance from the Belgrade Nikola Tesla airport to the artificial Lake Sava (source: Google Maps)*

Taking into account the fact that the aircraft noise is the highest as the aircraft flies overhead, one spot exactly beneath the expected flight trajectory has been chosen to perform measurement. Geographical coordinates of the measurement location were: 44°46'57.1"N 20°23'20.8"E. The location was suitable also because nearby there is a popular restaurant and a lot of people visit this place to relax.

2.3 Meteorological data

The measurement has been performed on Friday, September 16th 2016, from 10:15 a.m. to 2:15 p.m. The weather at the measurement location was calm, with very little wind and no clouds. Temperature was moving from 16°C in the morning up to 31°C in the afternoon.

Meteorological data for the Belgrade Nikola Tesla airport have been obtained from METAR reports for the observed period and they are shown in Table 2. The wind direction was variable and was moving between 180 and 230 degrees with reference to true north. The wind speed was moving between 2 and 7 knots which can be considered as light or gentle breeze.

Table 2 *METAR reports for the Belgrade Nikola Tesla airport (source: www.ogimet.com)*

METAR LYBE 161400Z 18005KT 130V220 CAVOK 30/14 Q1014 NOSIG=
METAR LYBE 161330Z 23006KT 200V270 CAVOK 30/14 Q1015 NOSIG=
METAR LYBE 161300Z 22007KT 160V300 CAVOK 31/12 Q1015 NOSIG=
METAR LYBE 161230Z 21005KT 160V260 CAVOK 30/13 Q1015 NOSIG=
METAR LYBE 161200Z 19005KT 130V260 CAVOK 30/14 Q1016 NOSIG=
METAR LYBE 161130Z 19004KT 110V230 CAVOK 30/14 Q1016 NOSIG=
METAR LYBE 161100Z VRB03KT CAVOK 29/15 Q1016 NOSIG=
METAR LYBE 161030Z 22003KT 170V280 CAVOK 29/15 Q1017 NOSIG=
METAR LYBE 161000Z VRB02KT CAVOK 28/15 Q1017 NOSIG=

2.4 Air traffic data

During measurement time, there were 22 departures and 29 arrivals at Belgrade Nikola Tesla airport. Due to meteorological situation and scheduled traffic at the airport, the runway in use was runway 12, which meant that aircraft were departing to southeast and were arriving from northwest. Therefore, we were able to record only departures at selected location. Flight information for actual departures from Belgrade Nikola Tesla Airport during the observed time is shown in Table 3.

Table 3 *Flight information for actual departures from Nikola Tesla Airport (source: www.beg.aero and Flightradar24)*

Airline	Flight number	Arriving airport	Aircraft type	Height	Lateral distance (m)	Time departed
Swiftair Hellas	MDF400P	Thessaloniki	Swearingen SA227AC	792m (2600ft)	442	11:03
Air Serbia	JU 380	London Heathrow	Airbus A319	914m (3000ft)	887	11:28
Ellinair	ELB 751	Heraklion	Boeing 737-300	1113m (3650ft)	1940	11:38
Etihad Airways	EY 72	Abu Dhabi	Airbus A320-200 sharklets	777m (2550ft)	10	12:40
Qatar Airways	QR 232	Doha	Airbus A320	831m (2725ft)	15	12:42
Air Serbia	JU 552	Istanbul Ataturk	Airbus A319	1036m (3400ft)	603	13:03
Air Serbia	JU 500	New York JFK	Airbus A330-200	671m (2200ft)	40	13:08
Lufthansa	LH 1407	Frankfurt	Airbus A319	1108m (3635ft)	55	13:10
Air Serbia	JU 122	Sofia	ATR 72	889m (2918ft)	1000	13:24
Air Serbia	JU 522	Thessaloniki	Airbus A319	1181m (3875ft)	55	13:30
Aeroflot	SU 2091	Moscow Sheremetyevo	Airbus A320	715m (2345ft)	1450	13:32
Air Serbia	JU 162	Skopje	ATR 72	852m (2794ft)	80	13:34
Lufthansa	LH 1723	Munich	Airbus A319	861m (2825ft)	770	13:37
Air Serbia	JU 142	Dubrovnik	ATR 72	709m (2325ft)	3820	13:40
Air Serbia	JU 152	Split	Airbus A319	1067m (3500ft)	45	13:46
Air Serbia	JU 224	Pula	ATR 72	975m (3200ft)	3300	13:49
Air Serbia	JU 512	Athens	Airbus A320	997m (3270ft)	30	13:52
Air Serbia	JU 182	Tivat	Airbus A319	823m (2700ft)	3350	13:56
Alitalia	AZ 591	Rome Fiumicino	Airbus A319	876m (2875ft)	1370	13:58
LOT Polish Airlines	LO 572	Warsaw	Embraer 170	926m (3037ft)	50	14:03
Air Serbia	JU 172	Podgorica	Airbus A319	853m (2800ft)	2100	14:07
Air Serbia	JU 122	Sarajevo	ATR 72	549m (1800ft)	3200	14:12

The data regarding airline, flight number, arriving airport, aircraft type and time of departure were downloaded from Nikola Tesla airport website, while height and lateral distance between aircraft and measurement location were determined based on radar data and geographical coordinates of the measurement locations. From Table 3 it can be seen that 22 departures were performed by ten different airlines operated with seven different aircraft types. The most frequent airline was Air Serbia with 13 departures, from which six were operated by Airbus A319, five by ATR 72, and the other two by Airbus A320 and Airbus A330-200.

Generally, airplanes could be divided into different categories based on the weight of the aircraft (light, medium, heavy). This should be taken into account because for higher Actual Take-off Weight (ATOW), aircraft will need a higher take-off speed and a longer take-off run which implies more noise on the ground since the aircraft will be closer to the airport surrounding during take-off. With the increase of ATOW, both the rate and angle of climb will be reduced which will significantly increase the noise level on the ground. In addition to the Operational Empty Weight (OEW) which is constant for one aircraft type, ATOW depends on payload (cargo, passenger, and luggage) and required amount of fuel that varies depending on the destination. For the purposes of comparison, the data regarding Maximum Take-off Weight (MTOW), number of seats, and ICAO Wake Turbulence Category (WTC) for seven aircraft types relevant for this study are shown in Table 4.

Radar data for every observed flight were obtained from Flightradar24 website (www.flightradar24.com) and they are graphically shown in Fig. 2.

Table 4 *Aircraft type data*

Aircraft type	MTOW[1] (t)	Number of seats[2]	WTC[3]
Airbus A319	64	128 - 153	M
Airbus A320	74	155 - 179	M
Airbus A320-200	74	150 - 179	M
Airbus A330-200	230	254	H
ATR 72	22	66 - 70	M
Boeing 737-300	56	148	M
Embraer 170	34	70	M
SA227AC Metro III SW4	7	18	L/M

[1] source: ICAO Doc. 7100 and http://www.flugzeuginfo.net
[2] source: http://www.flugzeuginfo.net and airlines fleet info from official websites
[3] ICAO Wake Turbulence Category, L=Light, M=Medium, H=Heavy

Fig. 2 *Radar data (source: www.flightradar24.com, using Google Earth for data presentation)*

From the Fig. 2 and Table 3, it can be seen that the majority of departures overfly the measurement location, except the very few due to the early right turn after take-off. Height ranges from 671m to 1181m which indicates significant differences between aircraft flight profiles. The reasons for that are above mentioned characteristics that influence aircraft performances during take-off.

2.5 Measurement equipment

All the equipment used for sound measurement and analysis was kindly provided by RMS d.o.o, who is the exclusive representative of the Danish firm "Brüel & Kjær" and the Danish-German firm "Brüel & Kjær Vibro" for Serbia, Macedonia, Montenegro and Republika Srpska (B&H).

For the measurements, two-channel hand held analyser type 2270 with standard accessories was used, including following software modules:

- sound level meter software, type BZ-7222
- real-time frequency analyser software, type BZ-7223
- logging software type, BZ-7224
- enhanced logging software – BZ-7225
- software for sound recording, BZ-7226.

For the analysis of recorded data, Brüel & Kjær's Measurement Partner Suite BZ-5503 was used.

3. RESULTS AND DISCUSSION

The aim of this set of measurements was to capture aircraft noise events and to determine the difference in noise levels between background noise and aircraft noise.

Apart from couple of measurements carried out to establish background noise levels, a total of 18 short-term measurements were done, one for each overflight. Even thought there were 22 departures during measurement time, only 18 of them were recorded since their flight trajectories were near measurement location. Due to the early right turn after take-off, for the other four departures, aircraft were too far away (more than 3km lateral distance) and there was no point to start measurement since their noise levels were below background noise.

Measurement results for background and aircraft overflight noise levels are shown in Table 5. Measurements are numbered from M1 to M19, where M1 represents background noise since there were no aircraft overflights during measurement time. Measurements from M2 to M19 represent 18 overflights that were recorded, sorted in ascending order by A-weighted Equivalent continuous sound level (LAeq). The results in Table 5 are divided into two parts. Overall results are related to the entire duration of each measurement that also included short periods before and after actual overflight, while event (overflight) results are related only to period where the event has been triggered manually or automatically by noise level exceeded. The last column in the table (ΔLAeq) represents the difference between LAeq of event, which in this case is aircraft overflight, and LAeq of M1, which is considered as a background noise level at given location. In addition to LAeq, indicators that were used in this study are: LAFmax and LAFmin, as a maximum and minimum level with A-weighted frequency response and Fast time constant; LAFn as a noise level exceeded for n% of the measurement period with A-weighted, calculated by statistical analysis - where n is between 0.01% and 99.99% (here five levels are shown LAF5, LAF10, LAF50, LAF90, and LAF95); and standard deviation.

Table 5 *Measurement results*

Measur-ement	Overall										Event (overflight)			ΔLAeq
	Duration	LAeq	LAFmax	LAFmin	LAF5.0	LAF10.0	LAF50.0	LAF90.0	LAF95.0	StdDev	Trigger	Duration	LAeq	
M1	64s	47.3	59.8	39.6	51.7	49.8	44.4	41.6	41.1	3.30				
M2	39s	58.8	66.2	49.9	63.4	62.4	57.2	52.4	51.7	3.71	-	-	-	11.5
M3	46s	59.2	66.2	49.7	63.6	62.5	58.5	52.6	51.4	3.78	Manual	39s	59.5	12.2
M4	55s	59.2	69.0	48.8	65.1	63.4	56.3	51.9	51.4	4.35	-	-	-	11.9
M5	61s	60.5	69.7	49.3	66.3	64.9	57.1	53.7	52.7	4.14	-	-	-	13.2
M6	55s	60.7	70.8	50.3	65.4	64.3	58.3	54.2	52.9	3.90	-	-	-	13.4
M7	90s	62.5	74.5	48.1	68.1	67.0	59.4	52.2	51.3	5.33	Manual	34s	64.7	17.4
M8	76s	62.9	72.2	49.8	68.2	67.3	59.6	53.1	52.2	5.19	Auto	23s	66.6	19.3
M9	65s	63.9	74.1	52.1	69.2	68.1	61.5	55.0	54.2	4.69	Auto	14s	67.9	20.6
M10	55s	63.9	72.1	49.3	68.7	67.7	61.8	53.1	51.2	5.23	Auto	15s	67.0	19.7
M11	62s	63.9	77.3	51.1	69.1	68.0	60.8	53.5	52.5	5.36	Auto	13s	66.8	19.5
M12	32s	63.9	76.3	54.2	69.7	67.3	59.8	56.4	56.0	4.06	Manual	25s	60.6	13.3
M13	55s	64.3	72.2	50.4	69.1	68.3	62.1	55.4	54.3	4.97	Auto	8s	67.2	19.9
M14	68s	64.6	73.9	47.1	70.7	69.6	58.7	50.3	49.3	7.36	Auto	20s	69.0	21.7
M15	65s	65.1	83.3	52.3	69.7	68.4	62.4	56.3	55.4	4.61	Auto	14s	68.2	20.9
M16	53s	65.2	74.4	54.9	70.5	69.3	61.9	57.5	56.7	4.47	Auto	16s	68.9	21.6
M17	54s	65.8	74.8	48.8	72.1	71.1	60.2	51.6	50.2	7.14	Auto	18s	69.9	22.6
M18	89s	70.1	82.1	47.3	77.9	76.2	57.8	50.2	49.3	9.65	Auto	25s	75.4	28.1
M19	64s	70.2	88.3	48.3	75.0	70.7	60.3	53.6	52.7	7.28	Auto	11s	68.9	21.6

3.1 Background noise level

In order to obtain background noise level at given location, several short-term measurements were conducted, all without the presence of aircraft noise. There was no need for longer continuous measurement, since all the measured noise levels were quite uniform during the entire period. Based on the noise levels from M1 measurement, LAeq = 47.3 dB was adopted as background noise level. LAF90 or LAF95 are also used as indicators of background noise levels while LAF10 or LAF5 are sometimes used to indicate the level of noise events. The values of LAF95 for 18 measurements (M2-M19) range from 49.3 dB to 56.7 dB with an average value of 52.2 dB which is higher than adopted background noise level but still relatively close considering some background events that occurred during overflights (dog barking, children screaming, all very near the microphone).

3.2 Aircraft overflights frequency and noise levels

The frequency of aircraft overflights is highly correlated with the annoyance. Based on the measurements, it was concluded that the minimum time between two overflights was 50 seconds, while maximum gap was 61 minutes with an average value of around 10 minutes between two overflights. The gap between the most of the overflights was between two and five minutes as it can be seen from Fig. 3.

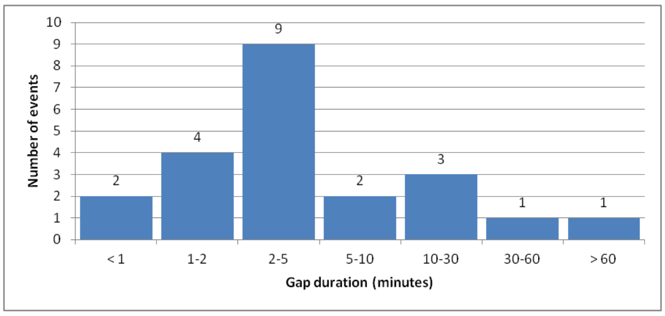

Fig. 3 *Gap between two events (overflights)*

From the Fig. 4 it can be seen that the most aircraft overflights occurred during the last half hour of the measurement period with an average time between two overflights of 3.75 minutes. On the other hand, there were two half-hour periods with no aircraft overflights.

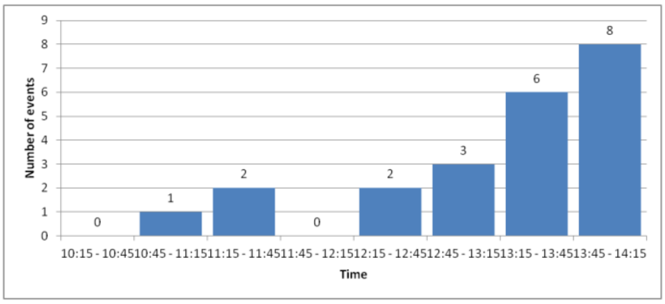

Fig. 4 *Aircraft overflight frequency*

Based on the measurement results presented in Table 5, from Fig. 5 it can be seen that for the most aircraft overflights (13 of them) LAeq ranges from 60 dB to 70 dB, while there are no overflights with LAeq less than 50 dB or higher than 80 dB. As expected, LAFmin does not exceed 60 dB for any overflight and for the most of them (11) is less than 50 dB. LAFmax exceed 80 dB during three overflights, while for 11 of them is between 70 dB and 80 dB.

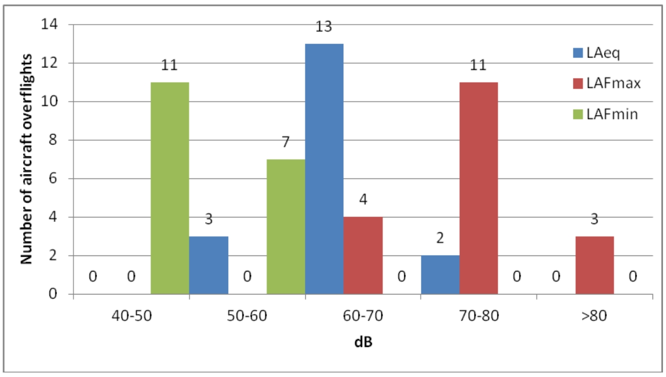

Fig. 5 *Clustering aircraft overflights according to LAeq, LAFmax and LAFmin noise indicators*

These levels refered to the entire period of each measurement which also included short periods before and after actual overflight.

In order to put a focus on aircraft overflights, level triggered marker was set for events above LAF = 70 dB(A) with rising start slope, start and stop duration of 2s and stop level of LAF = 65 dB(A). For each level triggered event, signal recording was automatically initiated which was very useful for later processing. Pre-recording and post-recording times were set to 10s and 2s, respectively with minimum duration limit of 10s. In addition to automatic signal recording, manual recordings were used when aircraft noise levels were below the trigger marker. Manually and automatically, level triggered event were determined for most of the overflights and their duration and LAeq level was presented in Table 5. These levels, together with previously adopted background noise level, were used to determine ΔLAeq that represents difference in noise levels in cases with and without the presence of aircraft noise. In four cases where event markers were not activated, the overall LAeq was used to compare with background noise level. From Table 5 it can be seen that ΔLAeq ranges from 11.5 dB for M2 to 28.1 dB for M18. Average ΔLAeq is 18.2 dB. It is well known that, due to the logarithmic scale, increase of 6 dB represents a doubling of the sound pressure and increase of 10 dB is considered to be significantly louder, as it is shown in Fig. 6.

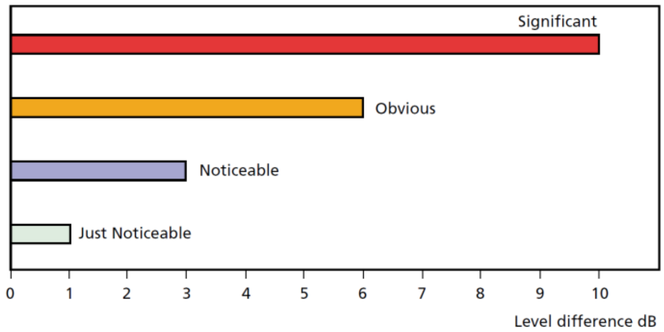

Fig. 6 *Perception of Sound* [7]

Having that in mind, level difference between background noise and aircraft overflights should be considered as significant.

3.3 Tone assessment

Noise with distinct tones, for example, noise from fans or compressors, is generally far more annoying than other types of noise [7]. In order to account for different perceptions and reactions of people to noise, the Rating Level (Lr) parameter has been developed to quantify noise annoyance in relation to the general population. The Rating Level is defined in the ISO 1996−2 standard [8], and among other thing, it consists of penalty for tones. The penalty for tones varies between 0 dB (no penalty) and 6 dB. Some countries use a single penalty value of 5 dB, while other countries use two or more steps [7]. The presence of tones could be determined subjectively, however in this study objective method based on 1/3-octave or FFT (Fast Fourier Transform) analysis has been used.

Tone assessment for each aircraft overflight measurement was done using Measurement Partner Suite BZ-5503. By comparing the tone level to the level of the surrounding spectral components, the presence of tone was found in measurement M17. From the Table 6 it can be seen that differences to the spectral components to the left and right are significant at tone frequency of 630 Hz and 2 kHz.

Table 6 *Pure tone assessment (measurement M17)*

Tone Frequency [Hz]	Level [dB]	Difference to the left [dB]	Difference to the right [dB]
630	64.3	6.5	6.3
2000	63.4	7.7	11.7

Significant difference between surrounding spectral components is easier noticeable on Fig. 7, while from measurement M17 profile shown in Fig. 8 it can be seen that the pure tone occurs in 22s of measurement when the aircraft noise is the highest (more than 70 dB).

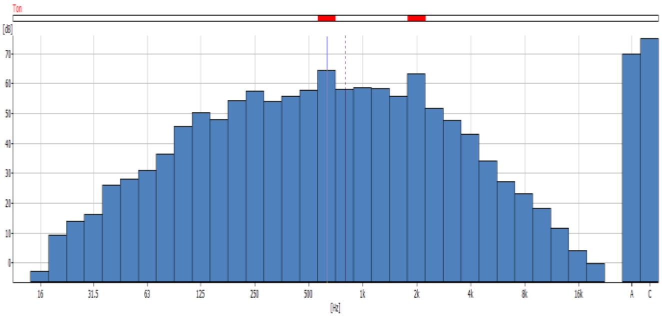

Fig. 7 *Pure tone assessment (measurement M17 spectrum)*

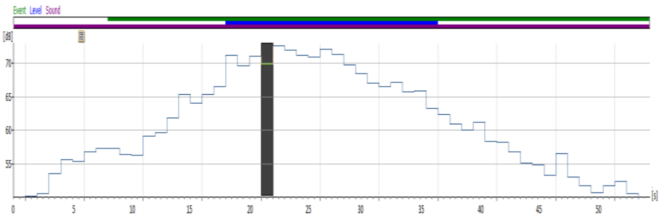

Fig. 8 *Pure tone assessment (measurement M17 profile)*

It should be taken into account that the presence of tones in measurement M17 was found only in several short intervals and not during the whole overflight. If the tone was present during the whole overflight period it would indicate that LAeq for this aircraft overflight should be increased by penalty value of 5 dB to account for noise annoyance which is not the case here.

3.4 Noise indicator limit values

Limit values for outdoor noise indicators in the Republic of Serbia, according to residential zones, are given in Table 7.

Table 7 *Limit values for outdoor noise indicators* [9], [10]

Zone	Land use	Noise level in dB	
		Day and evening	Night
1	For rest and recreation, hospitals and recovery facilities, cultural-historical locations, large parks	50	40
2	Touristic areas, camps and school zones	50	45
3	Residential areas	55	45
4	Business-residential areas, commercial-residential areas and children's playgrounds	60	50
5	City centre, trade, commercial, administrative zones with dwellings, areas along the motorways, main roads and city roads	65	55
6	Industrial, storage and servicing areas and transport terminals without dwellings	At this area borders, noise must not exceed the limit value of the neighbouring area	

Limit values are equal for day and evening time. They cover overall noise from all sources in the considered area.

The acoustic zoning for the city of Belgrade is not yet done. However, the area such as artificial Lake Sava should belong to the Zone 1 since it has large parks and the land use is for rest and recreation. It implies that L_{day} and $L_{evening}$ levels should not exceed 50 dB.

Even though this study is composed of several short-term measurements, it is possible to approximately calculate LAeq for the entire four-hour measurement period. Using the adopted background noise level for each gap between aircraft overflights (when there was no measurements) and the information about duration and time when overflights occurred, short-term measurements were transformed into four-hour measurement period as shown in Fig. 9. It was calculated that LAeq for the entire four-hour measurement period is 54.8 dB which is higher than limit value set for areas for leisure and recreation.

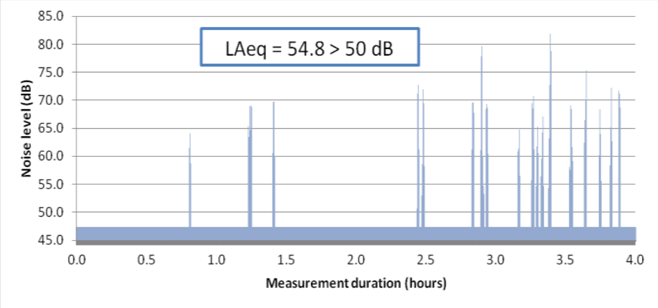

Fig. 9 *LAeq for the four-hour measurement period*

It should be taken into account that this value is based on only several short-term measurements and hypothesis that adopted background noise level was the same for each period without aircraft overflights. On the other hand, if the entire day and evening period of time (16 hours) was observed, 5 dB penalty for the evening period would increase the LAeq value since there are also a lot of aircraft overflights in that period of time.

4. CONCLUSION

Analysis presented in this paper has shown the influence of aircraft noise on quite areas. The difference in noise levels with and without aircraft noise was demonstrated on one example. Using relevant guidelines in the field of quiet area selection, the artificial Lake Sava located 9km from Belgrade Nikola Tesla airport has been chosen as a quiet area since it has required characteristics and frequent aircraft overflights. Several short-term measurements have been conducted at one location and the results were paired with the air traffic data. Background noise level has been determined which represented the noise levels during periods without aircraft overflights. During four hours, noise levels from 22 departures were measured and for majority of them LAeq ranged from 60 dB to 70 dB. Level difference between background noise and aircraft overflights ranged from 11.5 dB to 28.1 dB and proved to be significant. In addition, tone assessment showed the presence of tones in one measurement. However, it lasted only several short intervals and not during the whole overflight, and due to that LAeq should not be increased by penalty value of 5 dB to account for noise annoyance. Even though the acoustic zoning for the city of Belgrade is not yet done, approximately calculated LAeq for the entire four-hour measurement period showed that noise level could be higher than noise indicator limit value for Zone 1. However, this result should be taken with reserve since detailed measurements are needed in order to make more precise conclusion.

With the expected increase of air traffic on the Nikola Tesla airport, this area will certainly be more exposed to negative influence of aircraft noise, since it is located exactly beneath flight trajectories. One solution that could reduce noise exposure during departures from airport is to introduce mandatory right turn after take-off before reaching this quiet area. Based on the measurement results this action could be considered as successful.

In order to keep the aircraft noise on acceptable level and to protect quiet areas, continues noise and flight tracks monitoring is required and highly recommended.

ACKNOWLEDGEMENTS

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REFERENCES

[1] European Environment Agency, “Quiet areas in Europe - The environment unaffected by noise pollution,” 2016.

[2] A. Gidlöf-Gunnarsson and E. Öhrström, “Noise and well-being in urban residential environments: The potential role of perceived availability to nearby green areas,” *Landsc. Urban Plan.*, vol. 83, no. 2–3, pp. 115–126, Nov. 2007.

[3] D. Shepherd, D. Welch, K. Dirks, and D. McBride, “Do Quiet Areas Afford Greater Health-Related Quality of Life than Noisy Areas?,” *Int. J. Environ. Res. Public Health*, vol. 10, no. 4, pp. 1284–1303, Mar. 2013.

[4] EC, “Directive 2002/49/EC of the European parliament and the Council of 25 June 2002 relating to the assessment and management of environmental noise,” *Official Journal of the European Communities*, vol. 189, no. 12. pp. 12–25, 2002.

[5] “Transposition and implementation of environmental and climate change acquis - Chapter 27: status and plans.” Belgrade, Serbia, Sep-2015.

[6] European Environment Agency, “Good practice guide on quiet areas,” 2014.

[7] Brüel & Kjær, “Environmental Noise Booklet.” 2000.

[8] “ISO 1996-2 Acoustics - Description, assessment and measurement of environmental noise - Part 2: Determination of environmental noise levels.” 2007.

[9] Vlada Republike Srbije, “Uredba o indikatorima buke, graničnim vrednostima, metodama za ocenjivanje indikatora buke, uznemiravanja i štetnih efekata buke u životnoj sredini,” *Službeni Glas. RS*, vol. 75/2010, 2010.

[10] M. Praščević, D. Cvetković, D. Mihajlov, and N. Holoček, “The acoustic zoning - A comparasion of legislation and experiences in Italy and Serbia,” in *Proc. of the 4th International and 23rd National Conference Noise and Vibration*, 2012, pp. 21–28.

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EVALUATION OF NOISE POLLUTION BY STRATEGIC NOISE MAPS AND URBAN PLANNING

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Abstract - *Environmental noise pollution is a spatial phenomenon which can be quantified and graphically presented by strategic noise maps. For the assessment of population exposure to noise pollution the largest noise exposure database was created through Noise Observation and Information Service (N.O.I.S.E.) The result was that around 56 million people across EU are exposed to noise above 55 dB during daytime. The analysis of development process of cities in Serbia shows lack of studies concerning the issue of environmental noise pollution. Noise assessment by strategic noise maps is of great importance for land use planning for outlining the future location and type of land use development within urban areas. Strategic noise mapping allows researches and policy makers to identify the locations affected by excessive noise pollution. This paper intends to discuss the importance of strategic noise mapping for the assessment of noise pollution in urban environment and to introduce concept of noise mapping to urban designers and planners for future urban development.*

Keywords: *environmental noise pollution, strategic noise mapping, urban areas*

1. INTRODUCTION

Noise pollution is common problem of developed countries where the majority of population lives in urban areas exposed to noise which affects the quality of life and resident’s health and well-being. Life in the urban agglomerations generates sound that is usually described as unwanted sound – noise. The arising population density and the small distances between different areas of the city like residential, industrial and road surfaces have created a problem which is a subject of many environmental policies. [1] According to the World Health Organization (WHO) 40% of the EU population lives exposed to road traffic noise above 55 dB during the day and approximately 50% of Europeans live in zones of acoustical discomfort. [2] Basic principles of European noise policy are defined in Directive 2002/49/EC known as Environmental Noise Directive (END). [3] For environmental noise research, mapping is very important and it is required under the END Directive. Assessment of noise from major roads and in communities with more then 100 000 inhabitants is defined by END through two phases. The first phase regards to agglomeration with more than 250 000 inhabitants, all major airports, major roads and railways. The second phase includes agglomerations with more than 100 000 inhabitants and major roads. [3] Urban planning has strong potential for linking the noise mapping and environmental health. There are many factors that contribute to increased noise levels in cities. Population growth leads to high traffic volume. Spatially, roads are distributed closely to the areas where people work and live which leads to limited space and high rise buildings thus increasing the traffic volume. [2] Urban noise must be managed so that the excessive noise levels do not conflict with common human activities.

2. NOISE MAPPING

2.1. Methodology of noise mapping process

Noise mapping is an assessment method for the description of spatial distribution of noise levels. [3] This method allows an efficient visualization of noise distribution.

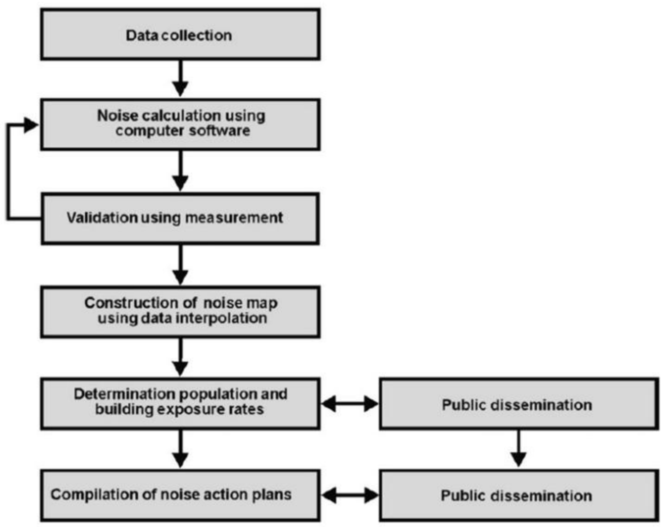

Fig 1. *Schematic overview of noise mapping* [2]

Strategic noise maps can be developed by calculating noise levels at receiver point on uniform grid over the study area using the various input data. Procedures for noise map creation are given on Fig 1. Input data include information about the terrain, land use, buildings and obstacles with heights, population data, and the data regarding noise sources such as traffic and industry (Fig 2).

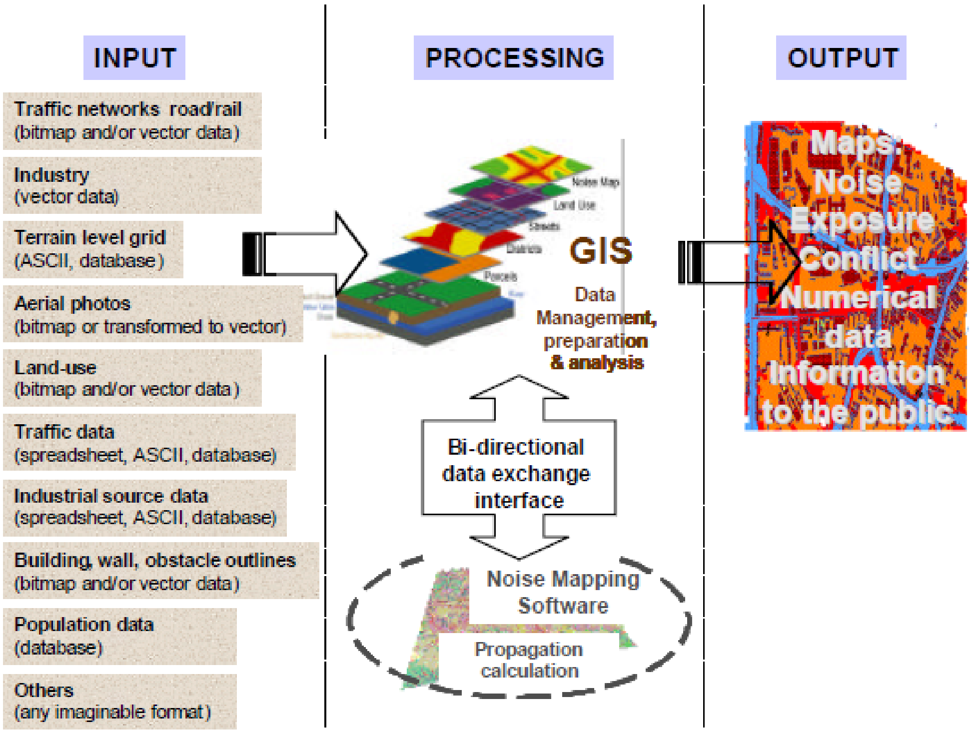

Fig 2. *Strategic noise mapping coupled with GIS* [4]

Additional information for each noise source include emission data (traffic density, types of vehicles, octave band spectra), propagation data (ground surface, road pavement, materials of the building's facade), meteorological data and other. [2] According to END, calculations are performed at receiver height of 4m above the ground. These calculations are usually undertaken using commercial software which includes algorithms for noise source propagation. [5]

After gathering the input data noise maps are then created using a spatial interpolation with GIS (Geographic Information System. The phenomena of noise involve spatial distribution and dynamic process that fits into GIS environment.

The noise emissions and propagation are calculated for each noise source (road, railway, air traffic, industry) and overlapped on the geographical information. Grid point distances vary between 5m and 30m, depending on the complexity of the area. Noise maps can be graphically presented as noise curves or zones with 5 dB intervals with appropriate color coding. (Fig 3.) Validation of modeled noise map must be carried out through measurements during some reference period. [3] The most common problem of data acquisition is road traffic data by means of quality (types of vehicles, time representativeness, and accuracy) and quantity (different road segments) required. [3]

Results from developing noise maps go in two directions. The first outcome is spatial distribution of noise levels on the basis of universal noise indicators L_{den} and L_{night} related to roads, railways, aircraft and industry and the second outcome of mapping process is calculation of population exposed to noise. Both outcomes correspond to diagnosis of current environmental situation and the necessary action plans that must be developed. [6]

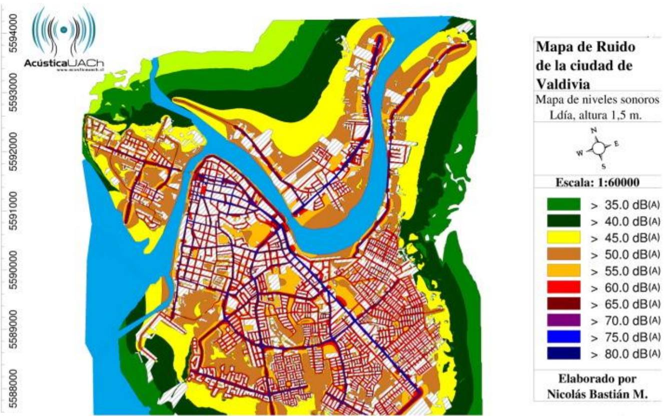

Fig 3. *Typical noise map showing the distribution of road traffic noise levels (city of Valdivia, L_{den})* [7]

3D noise maps are usually developed for high rise building severely affected by traffic noise for showing the influence of noise in all directions. [5] For policy makers and the members of public in Hong Kong 2D presentation of noise distribution was not good enough. In order to achieve more realistic presentation of noise exposure and better decision making researchers have developed 3D noise maps. Evaluation of noise pollution in high-rise build environment of Hong Kong is presented on Fig. 4.

(a) (b)

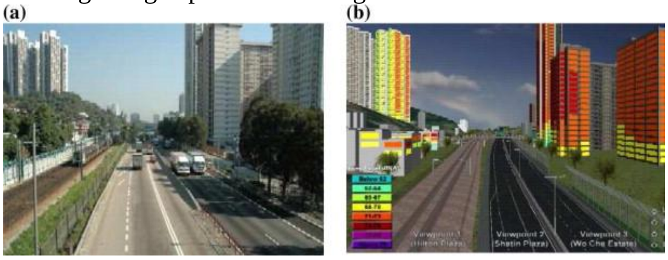

Fig 4. *Tai Po Road, (a) Photo. (b) Photorealistic 3D noise model with proposed noise barriers.* [8]

2.2. Noise conflict maps

Environmental acoustic zoning (noise zoning) is developed by municipalities to classify the land-use areas in the city by acoustic demands. Noise zoning is a part of urban environmental planning process. By overlapping noise maps with zoning maps, the difference between the predicted noise level and the noise limits of specific area can be marked as conflict zone. [6]

Conflict noise maps help authorities to identify so called “hot spots” – areas of excessive noise pollution as well as the “quiet areas”. Quiet areas inside urban zones represent an important value of the city life, as they offer relaxation from the everyday hustle and noise in a city. Such quiet areas can be city parks as well as inner gardens in housing blocks. Planners should design the city in a manner that certain areas retain quiet. The END demands that noise action plans shall include measures to preserve quiet areas. Quiet areas can include a range of different sites such as parks, residential areas, hospital areas, playgrounds, or cemeteries.

2.3. Noise action planning

END Directive is precise when it comes to the requirement of noise action plans. Action plans must include all necessary description of agglomeration all noise sources (roads, railways, airports etc.). A part of noise action plan is also the estimation of number of people exposed to noise and the identification of problems and situation that need to be improved. Actions with the competent authorities as well as the financial information (budget, cost-benefit assessment) are also part of these plans. [1] Local action plan aims to avoid noise, improve noise situation in areas where the noise exposure of residents is too high and to protect quiet areas and the recreational zones.

Kloth et al. propose different categories of measures for reduction of noise emission and exposure [1] :

To reduce noise at its source:
• low-noise road surfaces
• road traffic c management
• traffic calming
• low-noise tires
• low-noise vehicles
• driver behavior

To reduce the propagation of noise:
• land use planning and management
• noise screening
• buildings as noise barriers
• tunnels
• vegetation as noise shield

To reduce noise at the receiver:
• sound insulation
• building design

Measures for action planning can include changes in road or railway profile, low noise pavements (porous pavements), limitation or restriction of traffic but carefully chosen to meet inhabitants’ behavior by means of mobility or economical activities. Noise control devices such as acoustical barriers may be used in suburban highways but should be avoided within the city area because of their visual impact, light blocking etc. [3]

3. NOISE AND URBAN PLANNING

Spatial and urban planning outlines the future location and type of development activities within the regional and urban areas for a period of 5 to 15 years. It is a powerful tool for noise mitigation purposes. Competences concerning noise are often shared between local, regional and national authorities; their involvement can be crucial for the implementation of certain measures. Spatial planning has an impact on noise through defining the land use which impacts on traffic. [1] By spatial plans designers could allocate land use in such a manner to ensure distance between the noise source and the sensitive areas. If the quiet areas are identified by noise conflict maps spatial planners have the ability to preserve such areas against any new noise emissions. City developers through urban planning have the ability to promote public transportation, cycling and walking. If some areas of the city are part of the retrofitting process planners have the possibility of implementing noise abatement measures [2] Renewal of neighborhoods often includes redesign of roads, which can be used as an opportunity for noise issue consideration. [1] Development of new areas of the city increases traffic volume.

3.1. Recommendations for site and building location

Spatial distribution of noise pollution is affected by different urban design elements like terrain topography, presence of vegetation, street canyon design, distance from roads and orientation in relation to the noise source. Densely populated urban fabric is a mix of building, inhabitants, services and communications. Sound appears everywhere and urban planners and architects are not sufficiently aware of the interaction between the propagation of sound and the built outcome of their work.

Designing houses, relations between houses, connections between places, entire neighborhoods and cities is a serious task. [9] Useful tool in spatial planning that allows noise abatement is acoustical zoning. Best practice includes utilizing non sensitive buildings as noise barriers for sensitive building for example commercial buildings in front of a residential building to act like a barrier. (Fig. 5) [2] The propagation of noise can be reduced if appropriate types of buildings are put on the sound propagation path. A typical example includes the use of noise proofed terraces housing in the first raw facing the road instead of semi-detached housing in this way the front row would act like a barrier for semi-detached and detached houses behind as in the Fig. 6. [2]

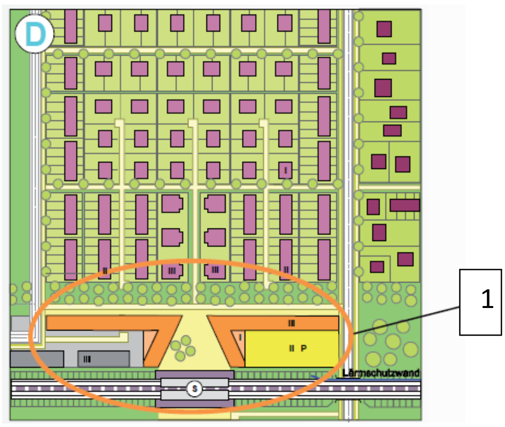

Fig. 5 *Buildings(1) as noise barriers along a railway line*[10]

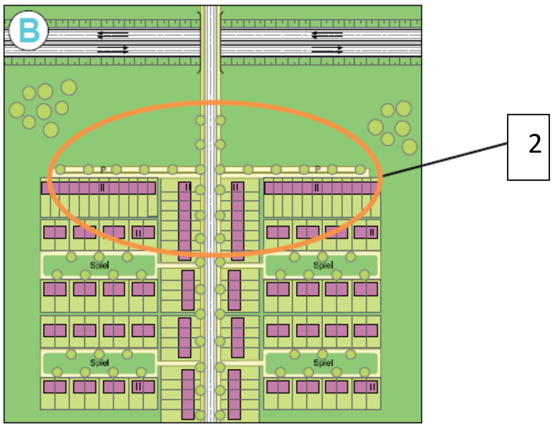

Fig. 6 *Closed front(2) of houses forms noise barrier* [10]

Acoustical shielding can be provided by the existing terrain, natural landscaping or wooded areas. A thoughtful chosen location and orientation of a building can positively influence on noise attenuation. From a noise perspective building geometry and orientation should be designed in such a way as to minimize potential reflections from key noise sources. Sites on rolling terrains separated from roads by trees and shrubs are quieter than those located on open ground.
Building wall which is facing the noise source should be windowless and thick and windows should be oriented to quiet areas of the site. The difference of sound pressure between these two orientations can be up to 5 dB. When it comes to windows they are marked as acoustically weak elements so the reduction of total window area is necessary. [11]

3.2. Recommendations for street canyon design for traffic noise reduction

Building shape, street geometry and the presence of urban furniture can have a strong impact on people's noise exposure. Sanchez et al. in their research [12] showed that building shape can be responsible for 7 dB variations in noise exposure at pedestrian level and 12-13 dB at building facade level. Further they concluded that the inclined ceiling or absorptive ceiling of balconies can reduce noise on upper stories for 13 dB. [12] (case F.3.3, F.3.4) In the case of general building shape (F1) upwardly inclined facades (F.1.3) are most efficient for noise reduction. Case F.4 showed that triangular prominences with up vertex are most efficient (F.4.5). In the case of F.5 concerning window inclination noise reduction is proportional to window inclination. [12]

4. SOUNDSCAPE

Numerous researches in the field of environmental noise mapping show that sound quality cannot be determined by a simple physical measurement, such as the usual A-weighted sound pressure level alone. Human perception of noise is not absolute and mainly relies on the meaning of sounds that is in relation to the sources emitting noise and the people who are exposed to it. [13] The word soundscape can then be used to describe the ensemble of sounds and noises that we experience in the environment where we live (or, better, in a specific part of that environment like a park, street, supermarket, airport, city center), over which we have little or no control, or a particular type of acoustical composition created by the artist to communicate a specific feeling. (5) Soundscapes that are mostly composed of traffic noise were described as unpleasant while ideal soundscapes were included a lot of human noises and were subcategorized according to the significance of the type of socialized activities performed and producing noises. [13]
The question is asked of how much noise is ‘‘too much’’ noise?
By following the soundscape approach researchers from Berlin won the European Soundscape Award 2012 for remodeling the city park in Berlin, Nauener Platz. (Fig 8.) Urban planners and acousticians reconstructed this park in order to improve ambiance and create more attractive acoustic environment. The project involved noise mapping, measurements, public discussions and workshops to make more pleasant environment [14-16]

5. CONCLUSION

Mitigation measures are essential to reduce or control the noise emitted from various activities in the urban centers. There is a need to develop acoustic design and planning guidelines for the reduction or elimination of unwanted sound. This paper intends to raise awareness among urban planners and architects that during the design phase of a new settlement or a refurbishment of an existing city area take into account the acoustic quality of the space.

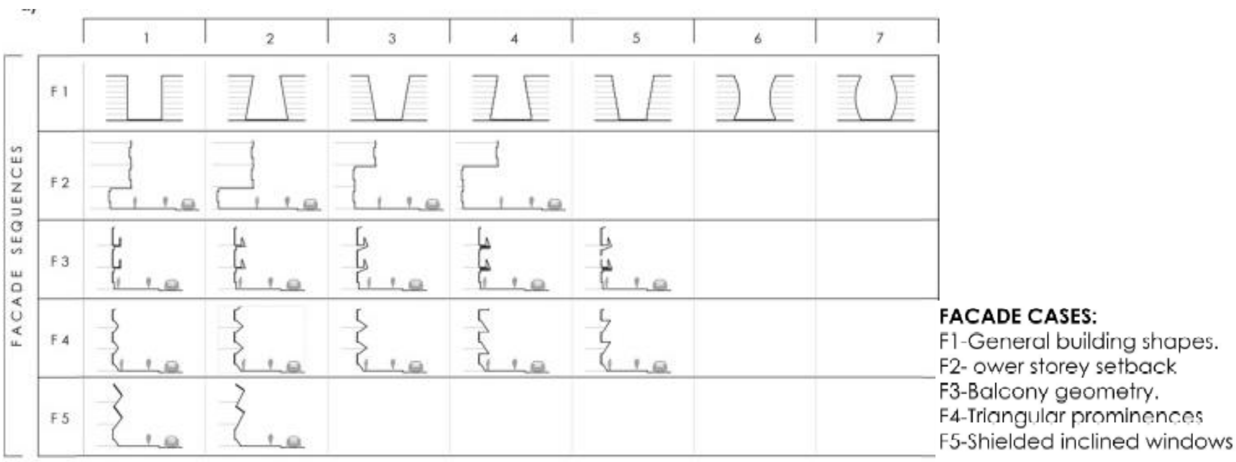

Fig 7. *Comparison of noise exposure along the street with different building geometries, adapted from* [12]

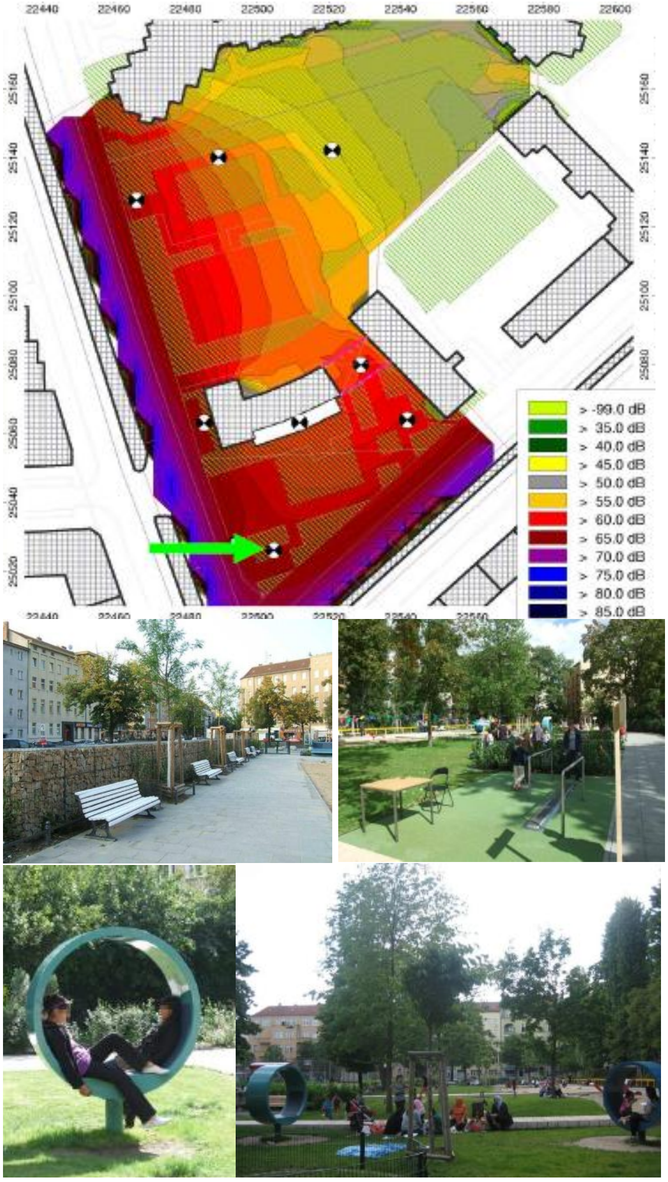

Fig 8. *Soundscape at Nauener Platz: noise mapping process, noise barrier (gabion wall) and playground* [14-16]

REFERENCES

[1] Murphy, E. & King, E., 2011. Environmental Noise Pollution, Elsevier Inc

[2] Coelho, J.L.B., 2002. On Noise Mapping and Noise Action Plans for Large Urban Areas. Forum American Bar Association, pp.1039–1044.

[3] Hällgren, N., 2012. Urban sound design – can we talk about it□? SoundEffects, 2(2), pp.36–50.

[4] European Commission Working Group Assessment of Exposure to Noise (WG-AEN), Good Practice Guide for Strategic Noise Mapping and the Production of Associated Data on Noise Exposure, 2006

[5] Kloth, M. et al., 2008. Practitioner Handbook for Local Noise Action Plans: Recommendations from the SILENCE project. , p.124. Available at: http://www.silence-ip.org/site/fileadmin/SP_J/E-learning/Planners/SILENCE_Handbook_Local_noise_action_plans.pdf.

[6] Kurakula, V., 2007. A GIS-Based Approach for 3D Noise Modeling Using 3D City Models. Earth.

[7] Nicolás A. Bastián-Monarca, Enrique Suárez, Jorge P. Arenas, Assessment of methods for simplified traffic noise mapping of small cities: Casework of the city of Valdivia, Chile, Science of The Total Environment, Volume 550, 15 April 2016, Pages 439-448, ISSN 0048-9697, http://dx.doi.org/10.1016/j.scitotenv.2016.01.139.

[8] Chi-wing Law, Chee-kwan Lee, Aaron Shiu-wai Lui, Maurice Kwok-leung Yeung, Kin-che Lam, Advancement of three-dimensional noise mapping in Hong Kong, Applied Acoustics, Volume 72, Issue 8, July 2011, Pages 534-543, ISSN 0003-682X, http://dx.doi.org/10.1016/j.apacoust.2011.02.003

[9] Licitra, G. & Als, E., 2012. Noise Mapping in the EU

[10] Melanie Kloth et. al. Practitioner Handbook for Local Noise Action Plans, framework of the SILENCE project on Quieter Surface Transport in Urban Areas, www.silence-ip.org

[11] Pandya, G.H., 2001. Urban Noise - A need for acoustic planning. Environmental Monitoring and Assessment, 67(3), pp.379–388.

[12] Gemma Maria Echevarria Sanchez, Timothy Van Renterghem, Pieter Thomas, Dick Botteldooren, The effect of street canyon design on traffic noise exposure along roads, Building and Environment, Volume 97, 15 February 2016, Pages 96-110, ISSN 0360-1323, http://dx.doi.org/10.1016/j.buildenv.2015.11.033.

[13] Raimbault, M. & Dubois, D., 2005. Urban soundscapes: Experiences and knowledge. Cities, 22(5), pp.339–350.

[14] Brigitte Schulte-Fortkamp and Pamela Jordan, When soundscape meets architecture, Noise Mapp. 2016; 3:216–231, DOI 10.1515/noise-2016-0015

[15] https://sonicagents.wordpress.com/2013/01/04/a-soundscape-re-modeled/

[16] http://www.eea.europa.eu/highlights/berlin-park-wins-award-for

EXPERIMENTAL RESEARCH TO ASSESS THE PERCEPTION OF CITIZENS ON NOISE POLLUTION, IN BRAILA, ROMANIA

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Abstract - *To carry out this study, a questionnaire was applied to a sample of 574 people in an area in Braila, Romania over a one-year period. The questionnaire contains 15 questions, as follows: a) the first questions focus on the identification data of the interviewed person, b) questions 7-10 and 13 refer to the main issues that concern people about noise pollution in Braila, c) questions 11-12 refer occupational groups that should be involved in problems of noise pollution, d) questions 14-15 relate to people's awareness of the current situation in noise pollution.*

In the second part of the paper is presented a correlation between the noise level and the stress level and a "map" of the average psychometric features of the subjects are drawn-up. Are taken in consideration the quasi-independent factors that allow the assessment of a person's stress: Stressor (noise); Cognitive intelligence quotient; Emotional intelligence quotient (Q.E.); intrinsic motivation; Degree of employability; Seniority and Speed of response.

The results show the dependency of the noise induced stress on subject age and gender, degree of education, but also on the season in which the experimental measurements were made.

1. INTRODUCTION

Today people live in a world of pollution. Pollution exists in various forms: pollution of air, water, soil, noise; there is garbage in cities, greenhouse effect, etc.

Each person reacts differently to the action of these factors, depending on age, sex, education, but also on their temperament.

Researches were made on people's perception of pollution: "the interrelated issues of urban sprawl, traffic congestion, noise, and air pollution are major socioeconomic problems faced by most European cities" [1]. Is very important "how social and cultural factors influence the way in which people interpret and make sense of risk, draws linkages with psychological risk perception research, revealing that over the last decade there has been a pronounced degree of convergence between the conclusions being reached across these two (historically disparate) fields of research" [2].

"Risk perceptions are important to the policy process because they inform individuals' preferences for government management of hazards that affect personal safety, public health, or ecological conditions. Studies of risk in the policy process have often focused on explicating the determinants of risk perceptions for highly salient, high consequence hazards" [3].

Maisonneuve, et al (2009) [4,5] presents "a new approach to monitor noise pollution involving citizens and built upon the notions of participatory sensing and citizen science", where it is suggested that "citizens to measure their personal exposure to noise in their everyday environment by using GPS-equipped mobile phones as noise sensors. The geo-localized measures and user-generated meta-data can be automatically sent and shared online with the public to contribute to the collective noise mapping of cities".

Urban noise and the implications it has on the population was also analyzed by Gidlöf-Gunnarsson and Öhrström (2007) [6]: "A questionnaire study was conducted in urban residential settings with high road-traffic noise exposure (L_{Aeq}, 24 h = 60–68 dB). Out of 500 residents, 367 lived in dwellings with access to a quiet side (L_{Aeq}, 24 h $\leq$ 45 dB free field value; "noise/quiet"-condition) and 133 had no access to a quiet side ("noise/noise"-condition). The present paper examines whether perceived availability to nearby green areas affects various aspects of well-being in these two noise-condition groups. For both those with and without access to a quiet side, the results show that "better" availability to nearby green areas is important for their well-being and daily behavior by reducing long-term noise annoyances and prevalence of stress-related psychosocial symptoms, and by increasing the use of spaces outdoors. In the process of planning health-promoting urban environments, it is essential to provide easy access to nearby green areas that can offer relief from environmental stress and opportunities for rest and relaxation, to strive for lower sound levels from road traffic, as well as to design "noise-free" sections indoors and outdoors [7, 8].

Unfortunately, people do not give sufficient attention to this extremely dangerous polluting factor: noise. For this reason, it may occur: hearing loss, presbycusis, occupational hearing loss, cardiovascular health problems, child physical development, and stress.

The mechanism of hearing loss can be attributed to aging, infection, surgery, prolonged use of some medications, trauma, and to stereocilia of the cochlea, the principal fluid filled structure of the inner ear.

The pinna combined with the middle ear amplifies sound pressure levels by a factor of twenty, so that extremely high sound pressure levels arrive in the cochlea, even from moderate atmospheric sound stimuli [9]. Presbycusis: The detection of high-pitched sound frequencies becomes more difficult. This affects speech perception, particularly of those words involving sibilants and fricatives. Both ears tend to be affected [10]. Hearing loss as a result of occupational exposure is one of the most common work-related illnesses. For example, musicians [11], miners [12] and those in manufacturing [13] and construction [14] may be exposed to higher and more constant noise levels.

Cardiovascular health problems: In 1999, the World Health Organization concluded that the available evidence suggested a weak correlation between long-term noise exposure above 67-70 dB and hypertension [15]. More recent studies have suggested that noise levels of 50 dB at night may also increase the risk of myocardial infarction by chronically elevating cortisol production [16, 17, 18].

Child physical development: The U.S. Environmental Protection Agency authored a pamphlet in 1978 that suggested a correlation between low-birthweight (using the World Health Organization definition of less than 2,500 g and high sound levels, and also high rates of birth defects in places where expectant mothers are exposed to elevated sound levels, such as typical airport environs. Specific birth abnormalities included harelip, cleft palate, and defects in the spine [19].

Stress: Research commissioned by Rockwool, a UK insulation manufacturer, reveals in the UK one third (33%) of victims of domestic disturbances claim loud parties have left them unable to sleep or made them stressed in the last two years. Around one in eleven (9%) of those affected by domestic disturbances claims it has left them continually disturbed and stressed [20]. More than 1.8 million people claim noisy neighbors have made their life a misery and they cannot enjoy their own homes. The impact of noise on health is potentially a significant problem across the UK given that more than 17.5 million Britons (38%) have been disturbed by the inhabitants of neighboring properties in the last two years. For almost one in ten (7%) Britons this is a regular occurrence [21].

Noise, but also its perception was analyzed by the author in several separate cases [22, 23, 24, 25, 26].

2. MATERIALS AND METHODS

This work analyses the perception of citizens on noise pollution, in a crowded area of the city of Braila, Romania, using specialized questionnaires, and the results were used to shape the psychological profile of the subjects.

To carry out this study, a questionnaire was applied to a sample of 574 people in a noisy area in Braila, Romania over a one-year period. The questionnaire contains 15 questions, as follows:

a) the first questions focus on the identification data of the interviewed person,

b) questions 7-10 and 13 refer to the main issues that concern people about noise pollution in Braila,

c) questions 11-12 refer occupational groups that should be involved in problems of noise pollution,

d) questions 14-15 relate to people's awareness of the current situation in noise pollution.

In the second part of the paper is presented a correlation between the noise level and the stress level and a “map” of the average psychometric features of the subjects is drawn-up.

The quasi-independent factors (on a scale from 0 to 100) which allow the stress assessment of a person, are:

1. Stressor (noise),
2. Cognitive intelligence quotient (Q.I.),
3. Emotional intelligence quotient (Q.E.),
4. Intrinsic motivation,
5. Degree of employability,
6. Seniority,
7. Speed of response.

Analyzing the subject’s traits "map" will be possible to characterize the stress level. Thus it will be seen if the subject lives in a harmful environment, has a standard intelligence, has an optimal level of emotional intelligence, has a relatively low intrinsic motivation, integrates within specific professional knowledge, according to the experience gained in his activity, his reaction speed is maximal and has an average cultural horizon.

In terms of psychometric assessment, all the factors mentioned above can be evaluated and quantified. Based on the development of an expected model of the powerful decision maker’s "map", one can highlight the differences and then make predictive analysis.

3. RESULTS AND DISCUSSION

Over a year 574 persons were interviewed in terms of pollution – pollution in general and noise pollution, in particular. Of these 67.82% are women.

Fig. 2 and 3 show that most people have average training or are retired and Fig. 4 reveals that the main concern of the subjects is way environmental pollution affects their health state.

The main issues that concern people about environmental pollution (Fig. 5) are air pollution and trash; noise pollution is not among the top concerns of citizens (Fig. 6). In fact the measured noise level is extremely high (Fig. 7).

During the analyzed period, the obtained average is 68.6 dB in the morning respectively 87.2 dB in afternoon. These figures far exceed the threshold accepted by the law. Fig. 8 summarizes the averages of the results obtained from the subject’s responses to the psychological questionnaires, from which was made an assessment of how citizens perceive noise.

Tab. 1 shows that all the sides contained in the psychological questionnaires attain a minimum during the summer months (holiday) and a maximum during the colder months. For example, in November, compared to August, stress due to noise is 100% higher Q.I. is higher by 70%, Q.E. is higher by 100%, intrinsic motivation is higher by 62.5% and speed of response increased by 40% in the same conditions of employability degree and seniority.

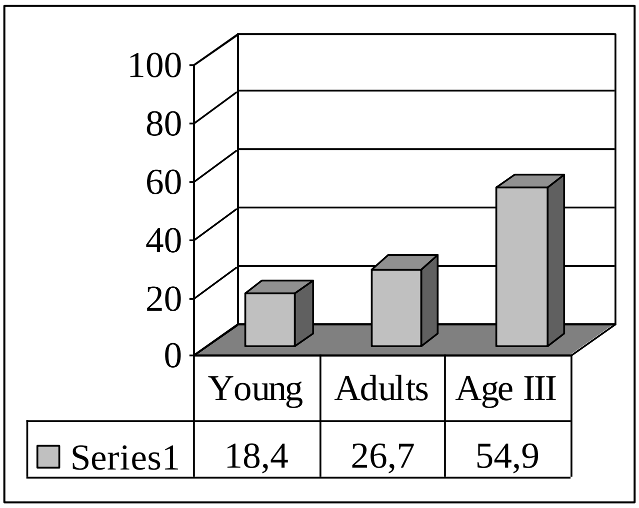


Fig. 1 *State of the surveyed persons – by age (Percentage of people interviewed)*

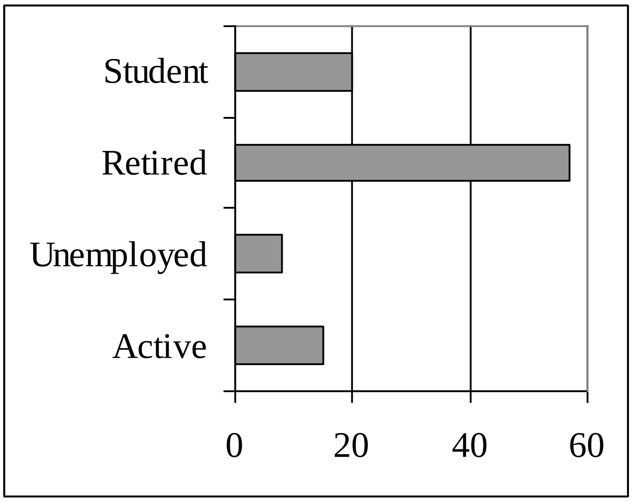


Fig. 3 *State of the surveyed persons – by the level of activity (Percentage of people interviewed)*

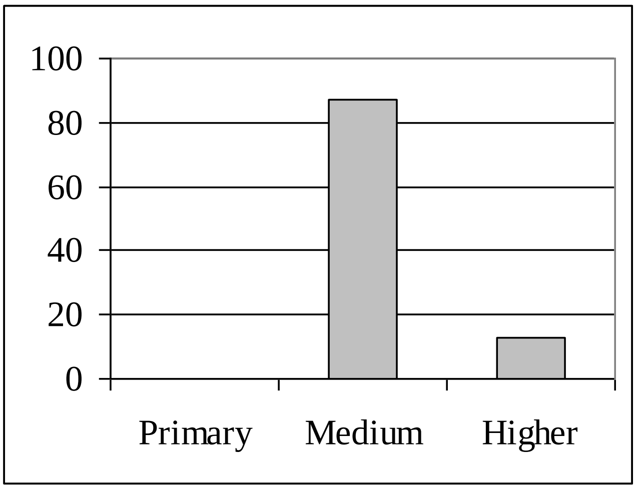


Fig. 2 *State of the surveyed persons – by training level (Percentage of people interviewed)*

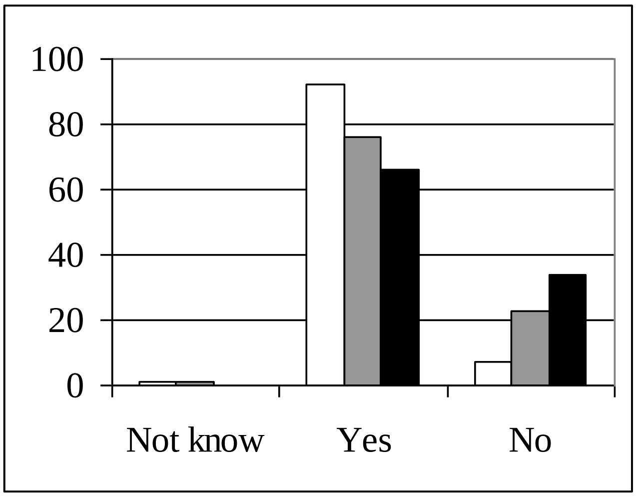


Fig. 4 *Environmental status influences the health of the population (Percentage of people interviewed)*

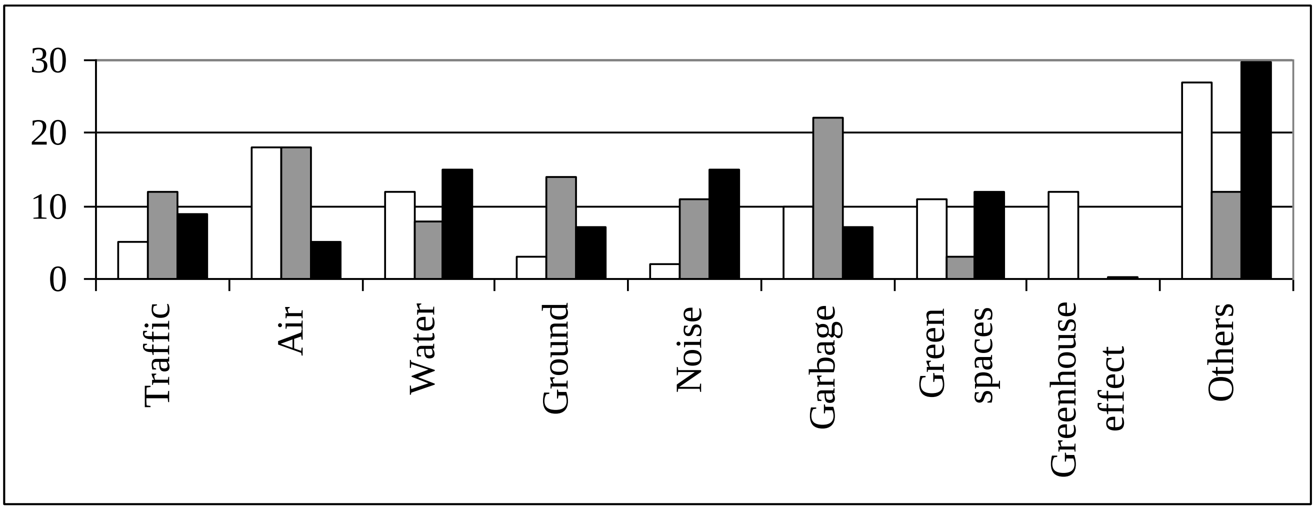


Fig. 5 *The main issues that concern people about environmental pollution (Percentage of people interviewed)*

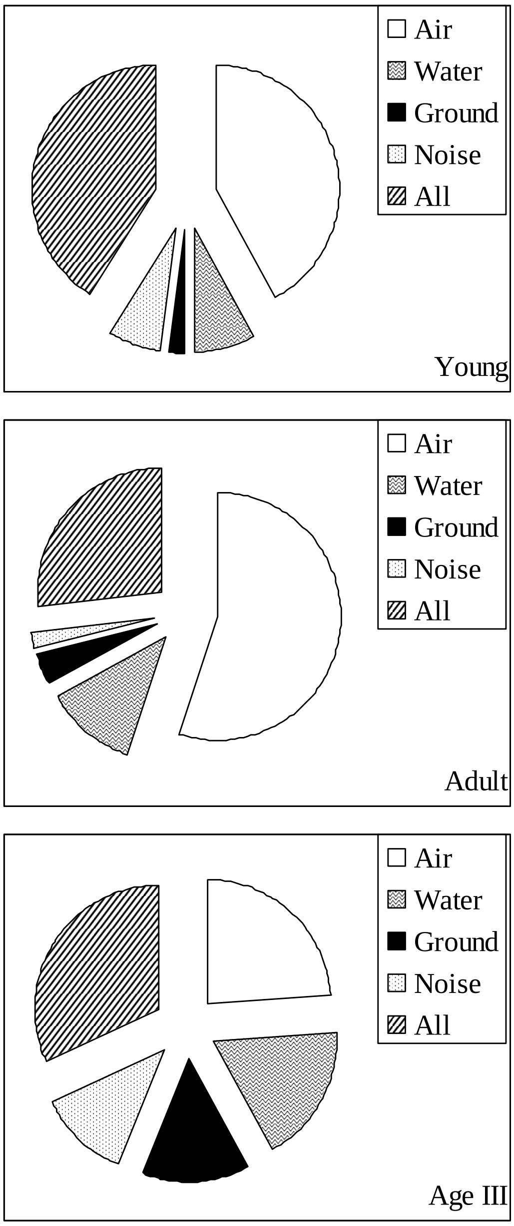


Fig. 6 *Environmental component considered as the most polluted (Percentage of people interviewed)*

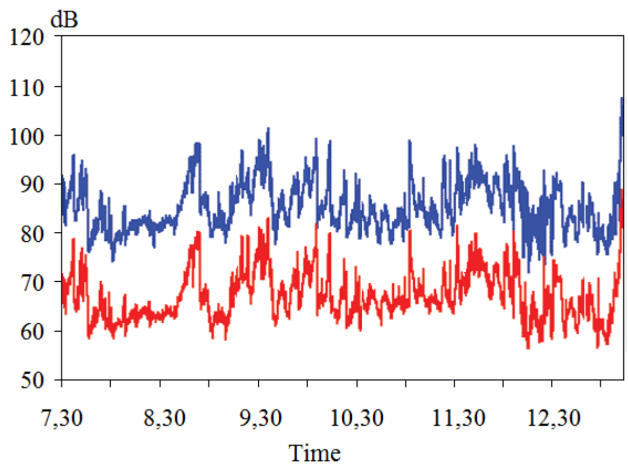


Fig. 7 *Level variation measured in the considered area: November 15, 2015 (red – morning); (blue – afternoon)*

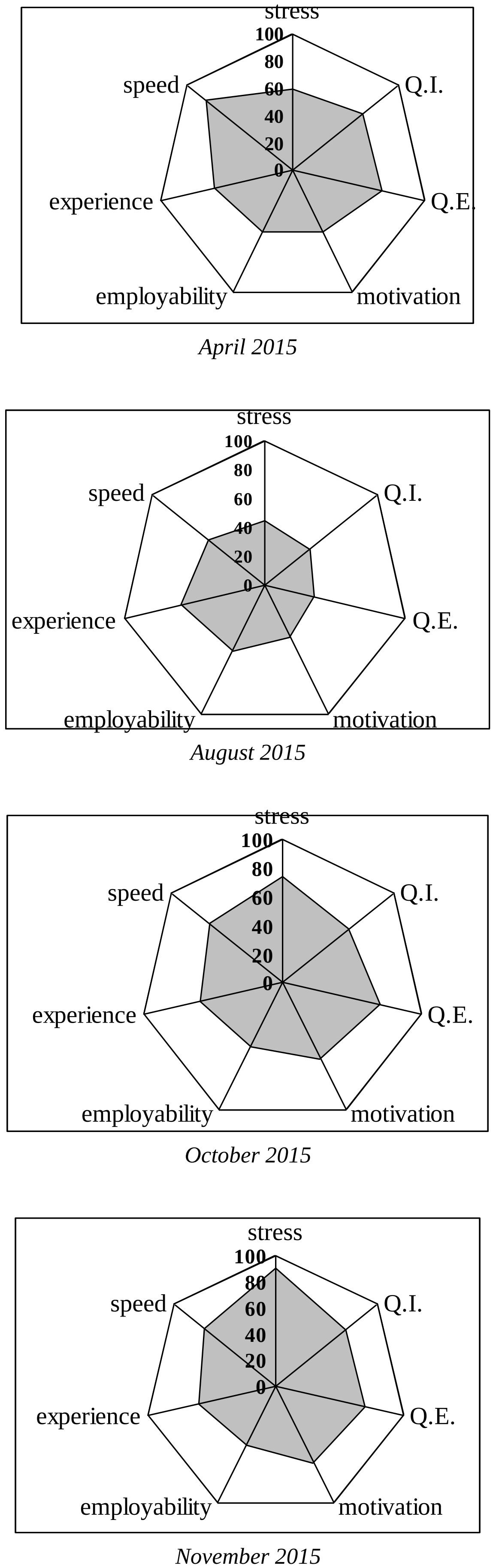


Fig. 8 *"Map" of the average psychometric features of the subjects*

Table 1 *The results obtained from the subject's responses to the psychological questionnaires*

	Stressor (noise)	Cognitive intelligence quotient (Q.I.)	Emotional intelligence quotient (Q.E.)	Intrinsic motivation	Degree of employability	Seniority	Speed of response
apr	60	66	68	50	50	60	82
aug	45	40	35	40	50	60	50
oct	74	60	70	60	50	60	65
nov	90	68	70	65	50	60	70

A classic example is the way in which the population has felt the noise stress made by the music from an outdoor restaurant, received outdoors (Fig. 9). This noise is about the same throughout the year; the only difference is the way people perceive this noise.

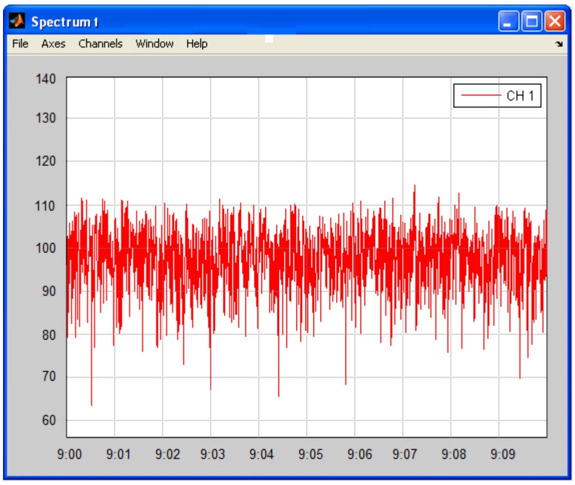


Fig. 9 *The noise made by the music in restaurant received outdoors*

Fig. 10 shows that – for the period studied – the population stress has a parabolic variation given by Eq. (1) where M = Month.

$$\text{Stress} = 7.1964M^2 - 50.089M + 135.5 \quad (R^2 = 0.927) \qquad (1)$$

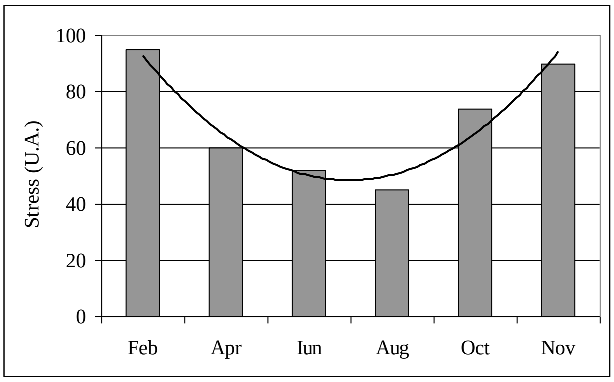
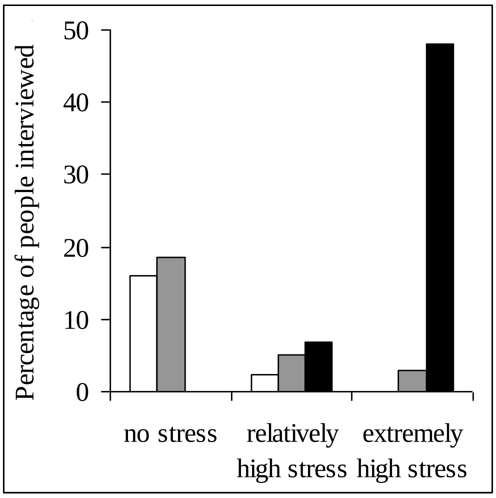


Fig. 10 *Variation of population stress in areas where measurements were made due to acoustic pollution for the analyzed months (2015)*

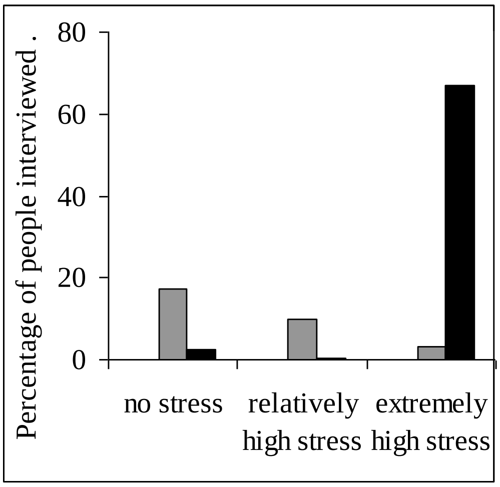
4. CONCLUSIONS

Regarding the perception of citizens on noise pollution, in Braila, Romania, is highlighted once again that people perceive stressors less in summer, during holidays and more in colder periods. If in the summer months, people accept noise easier, in colder months the noise is much more stressful.



by age

(□) - young; (■) - adults; (■) - age III



by training

(■) - medium; (■) - higher

Fig. 11 *The percentage of people under stress due to noise*

Also, the results show the dependency of the noise induced stress on subject age and degree of education: older people and people with higher education feel the noise harder than younger ones and those with secondary education (Fig. 10, 11).

Most interviewees pointed out that the population is not informed about the health hazards that may occur due to exposure to noise, most information came only from scientific broadcasts featured on TV) (Fig. 12), but these programs are not enough watched, so people do not know the risks they are expose to.

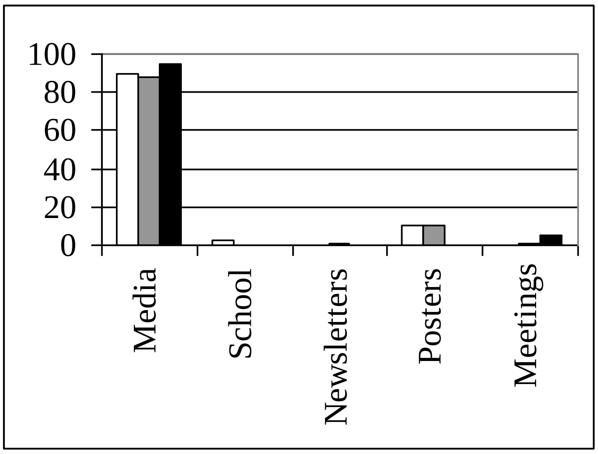


Fig. 12 *The main ways of information (Percentage of people interviewed)*
(□) - young; (■) - adults; (■) - age III

In conclusion, we can say that for the studied area, the noise level is well above the limit of noise, but people perceive it differently according to gender, age, education and season. Unfortunately, there is not education of citizens regarding the health hazards due to noise and for this reason, many young, later adults, suffer from hearing problems.

REFERENCES

[1] De Ridder K., Adamec V., Bañuelos A., Bruse M., Bürger M., Damsgaard O., Dufek J., Hirsch J., Lefebre F., Pérez-Lacorzana J.M., Thierry A., Weber C., “An integrated methodology to assess the benefits of urban green space”, *Science of The Total Environment*, Vol. 334–335, pp 489–497, 2004

[2] Bickerstaff K., **“**Risk perception research: socio-cultural perspectives on the public experience of air pollution”, *Environment International*, Vol. 30, Issue 6, pp. 827–840, 2004

[3] Gerber B.J., Neeley G.W., “Perceived Risk and Citizen Preferences for Governmental Management of Routine Hazards, Public Policy & Administration”, *Policy Studies Journal*, Vol. 33, Issue 3, 2005

[4] Maisonneuve N., Stevens M., Niessen M.E., Hanappe P., Steels L., “Citizen noise pollution monitoring”, Proceedings of the 10th Annual International Conference on Digital Government Research: Social Networks: Making Connections between Citizens, Data and Government, pp. 96-103, 2009

[5] Maisonneuve N., Stevens M., Niessen M.E., Hanappe P., Steels L., “Measuring and mapping noise pollution with mobile phones”, *Information Technologies in Environmental Engineering*, Part of the series Environmental Science and Engineering, pp 215-228, 2009

[6] Gidlöf-Gunnarsson A., Öhrström E., “Noise and well-being in urban residential environments: The potential role of perceived availability to nearby green areas”, *Landscape and Urban Planning*, Vol. 83, Issues 2–3, 19, pp 115–126, 2007

[7] Semenza J.C., Wilson D.J., Parra J., Bontempo B.D., Hart M., Sailor D.J., George L.A., “Public perception and behavior change in relationship to hot weather and air pollution”, *Environmental Research*, Vol. 107, Issue 3, pp. 401–411, 2008

[8] Trombetta Zannin P.H., Coelho Ferreira A.M., Szeremetta B., „Evaluation of Noise Pollution in Urban Parks”, *Environmental Monitoring and Assessment*, Vol. 118, Issue 1, pp. 423-433, 2006

[9] Henderson D., Bielefeld E.C., Harris K.C., Hu B.H., “The role of oxidative stress in noise-induced hearing loss”, *Ear Hear*, Vol. 27, Issue 1, pp 1–19, 2006

[10] Huang Q., Tang J., “Age-related hearing loss or presbycusis”, *European Archives of Oto-Rhino-Laryngology*, Vol. 267, Issue 8, pp 1179–1191, 2010

[11] Kardous C., Themann C., Morata T., Reynolds J., Afanuh S., “Reducing the risk of hearing disorders among musicians”, Cincinnati, OH: U.S. Department of Health and Human Services, Centers for Disease Control and Prevention, National Institute for Occupational Safety and Health. DHHS (NIOSH) Publication No. 2015–184, 2015

[12] “Hearing Loss Prevention”, Centers for Disease Control and Prevention, *The National Institute for Occupational Safety and Health* (NIOSH), Retrieved July 12, 2016, http://www.cdc.gov/niosh/mining/topics/HearingLossPreventionOverview.html

[13] “Occupationally-Induced Hearing Loss: Employers and Employees in Manufacturing need your help”, *National Institute for Occupational Safety and Health*, March 2010, http://www.cdc.gov/niosh/docs/2010-136/pdfs/-2010-136.pdf

[14] "Worker Safety Series: Protecting Yourself from Noise in Construction". *Occupational Safety and Health Administration*, 2016. https://www.osha.gov/Publications/3498noise-in-construction-pocket-guide.pdf

[15] Berglund B., Lindvall T., Schwela D., Goh K.T., "World Health Organization: Guidelines for Community Noise", *World Health Organization*, 1999. http://www.who.int/docstore/peh/noise/guidelines2.html

[16] Maschke C., Harder J., Ising H., Hecht K. and Thierfelder W., “Stress Hormone Changes in Persons exposed to Simulated Night Noise”, *Noise Health*, Vol. 5, Issue 17, pp 35–45, 2002.

[17] Franssen E., van Wiechen C.M.A.G., Nagelkerke N., and Lebret E., „Aircraft noise around a large international airport and its impact on general health and medication use”, *Occup Environ Med*, Vol. 61, Issue 5, pp 405–413, May 2004

[18] Lercher P., Hörtnagl J. and Kofler W.W., "Work noise annoyance and blood pressure: combined effects with stressful working conditions", *Int Arch Occup Environ Health*, Vol. 65, Issue 1, pp 23–28, 1993

[19] *** "Noise: A Health Problem" United States Environmental Protection Agency, *Office of Noise Abatement and Control*, Washington, D.C. 20460, August, 1978

[20] Cinderby, S., Snell, C., Forrester, J., "Participatory GIS and its application in governance: the example of air quality and the implications for noise pollution", *Local Environment: The International Journal of Justice and Sustainability*, Vol. 13, Issue 4, pp. 309-320, 2008

[21] Clout L., "How Noisy Neighbours Blight Millions of Lives", *The Daily Express*, Apr 9, 2009

[22] Picu, M., "Noise impact assessment using noise maps and noise conflict maps in Braila – A city situated between the largest RAMSAR Romanian areas: Danube Delta and Braila Small Island", *Carpathian Journal of Earth and Environmental Sciences*, February 2016, Vol. 11, No. 1, pp. 105-112, 2015

[23] Picu, M., "Multi-stress and human performance: a refutation of inverted-U hypothesis", *Journal of Multidisciplinary Engineering Science and Technology* (JMEST) ISSN: 3159-0040, Vol. 2, Issue 9, Sept, 2015

[24] Picu, M., "Noise study of a naval ship in different operating modes on the Danube", *Annals of Dunarea de Jos University Of Galati*, Fascicle XIV, Mechanical Engineering, pp. 55-62, 2014

[25] Picu, M., "Traffic, railway and industrial noise – the discomfort reflected in conflict maps. A case study in Braila, Romania", *International Workshop "Environmental Conflicts: Planning, Issues and Resolution",* Centre for Environmental Research and Impact Studies, Bucharest, September 4–6, 2014

[26] Picu, M., "Paradigm changing of acoustic comfort. Case study: noise pollution in two extremely populated areas in Braila", *Annals of the University Dunarea de Jos of Galati*, Fascicle II, Year V(XXXVI), No. 2, pp. 432-440, Mathematics, Physics, Theoretical Mechanics, 2013

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REDUCING THE NOISE FROM AN AIR CONDITIONING INSTALLATION: A CASE STUDY

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Abstract - *Air conditioning in swimming pools located in confined spaces is made using appropriate equipment. In some cases these are located close to residential areas and noise generated by air conditioning installations has high levels affecting the peace and health of people in these areas. The paper presents an investigation of the noise generated by such a facility identifying the sources of noise levels and characteristic spectra, ways of spreading, harmful effects, setting reduction methods, their application and their efficiency analysis.*

1. INTRODUCTION

There exist situations where air conditioning installations from some closed spaces involving swimming pools are placed in the vicinity of residential areas. Such installations generate an acoustic field characterized by a high noise level, with spectral components which affect the life of inhabitants from these areas.

In order to avoid such a situation, we developed in this paper an investigation of the acoustic field generated by such a noisy air conditioning installation by identifying the main sources of noise, noise levels and characteristic spectra, noxious effects, admissible limits and propagation way.

As a results of these investigations, there were established and implemented appropriate methods intended to reduce the noise. The efficiency of these methods has been also analysed.

2. NOISE SOURCES FROM THE AIR CONDITIONING INSTALLATION

The air conditioning installation which serves the closed space where the swimming pool is placed is located in the vicinity of a residential area, the distance to the nearest building being 32 m.

The parts of the installation are placed on a special place having the dimensions 4.5 m x 2.5 m x 2 m and are mounted on a concrete foundation through mounting elements having unfortunately poor damping properties.

The installation has a continuous functioning, being used for circulating the air from the closed space of the swimming pool.

Taking into account the constructive realization of the installation and its functioning way, the main sources of noise are ventilators, air ducts, bends, ramifications, adjusting valves, air vents for discharge and suction, engines of ventilators and pumps. During the functioning of the installation, these sources are vibrating. Their vibrations are transmitted to the air and then, of nearby to the entire environment as acoustic waves.

Because of the fact that these sources have different forms, the waves which are formed could be spherical, cylindrical and, at long distance, even plane waves. Taking into account the number, the shape and the size of the perturbing sources, as well as the fact that the reflected waves are overlapped on the direct ones, the resulted acoustic field is very complex.

The acoustic field generated by the air conditioning installation can be characterized by the level of acoustic pressure given by

$$L_p = 10\lg\left(\frac{p}{p_0}\right)^2 \tag{1}$$

where p is the effective acoustic pressure expressed in Pa and p_0=20 μPa, the reference acoustic pressure. The A-weighted continuous equivalent level of pressure in dB will be

$$L_{Aeq,T} = 10\lg\left[\frac{1}{t_2 - t_1}\int_{t_1}^{t_2}\frac{p_A^2(t)}{p_0^2}dt\right] \tag{2}$$

determined in an interval of time which starts at t_1 and ends at t_2, while another important characteristic indicator would be the percentage level $L_{AN,T}$, which represents the level of the weighted acoustic pressure obtained using the time constant "F" (fast) which is exceeded in N% of the considered time.

If within the Eq.(1) the acoustic pressure is A-weighted, then consequently we have the A-weighted acoustic pressure level L_{pA}.

These acoustic pressure levels could be obtained by computations using Eqs.(1) and (2) or could be directly measured using for example the noise analyzer B&K 2250, which facilitate the measurement and recording of the main characteristic parameters of the noise, as well as the spectral analysis in frequency bands of 1/1 or 1/3 octave or the statistical distribution of the noise level.

3. NOXIOUS EFFECTS OF THE NOISE GENERATED BY THE AIR CONDITIONING INSTALLATION

The noise generated by the sources from the air conditioning installation has noxious effects on the life and health of people leaving in the neighbourhood.

Having a continuous functioning, the generated noise is disturbing during the whole day, when it is adding up to the noise produced by the road traffic, which in the considered situation is very intense, on a road recently modernized. This noise is also disturbing during the night, affecting the desired quiet necessary for a good sleep.

Because of the presence in the considered area of high buildings on both sides of the road, it is obvious that this road receives clear characteristics of canion street, where the acoustic waves suffer multiple reflections which lead to an amplification and persistence of the noise.

The noise affects the nervous system of the inhabitants from neighbourhood generating psychophysiological modifications, blood circulation modifications, sleep disturbances. It also influence the visual function and the functioning of the endocrine glands, generating biochimical disturbances.

Under the action of a high noise level, the blood pressure raises, the pulse is accelerating, the intracranial vascular tension can be increased by 3 times, the visual acuity decreases and the rhythm of breath is changing.

The noise also produce nervous irritation, the fatigue process becomes pronounced, the attention and psychical reaction are lowered. Astenia and even nervous diseases could appear.

Exposure to a high level noise produce a lowering of auditory capabilities and could produce even sonorous trauma.

The action of the noise during the night on the inhabitants generate disturbance in the resting period contributing to the raising of the general fatigue degree of the human body lowering in this way the work yield.

In order to prevent and diminish these damaging effects of the noise there are established admissible limits of the equivalent noise level and of the noise curves in the vicinity of residence buildings and inside them.

In this respect, according to the Romanian standard STAS 10009-88, "Urban Acoustics", the positioning of residence buildings on streets of different technical categories or at the limit of some functional zones, as well as organizing of the road traffic must ba made so that to ensure the value of 50 dB(A) for the equivalent noise level measured at 2 m in the fromt of the building acording to another Romanian standard, STAS 6161-1:2008 and the spectral components cannot exceed the noise curve Cz45.

Moreover, the Romanian standard STAS 8156-86 "Protection against the noise in civil and socio-cultural buildings" establishes the admissible limits for the equivalent noise levels inside the civil and socio-cultural buildings due to exterior noise sources. Thus, for the apartments from residence buildings, the admitted limit of the indoor noise level is 35 dB(A) and the noise curve Cz30.

Besides, the ordinance of the Romanian ministry of health no. 536/1997 concerning the living environment of people establishes that these values admitted for the exterior and interior of residential buildings must be reduced during the night with 10 dB(A) as against the values admitted during the day.

By investigating the acoustic field generated by the air conditioning installation, one can find if it generate a comfortable or uncomfortable living environment.

In case if the noise level exceeds the admissible values, one can be established and implemented measures for reducing this noise.

4. INVESTIGATION OF THE ACOUSTIC FIELD

As we have mentioned, the acoustic field generated by the sources within the air conditioning installation serving the swimming pool is very complex. The sources generate spherical, cylindrical and plane waves. The acoustic pressure in a point of the field is obtained by adding acoustic pressures corresponding to each type of wave.

In the case of propagation of perturbation as spherical wave in an elastic, homogenous and isotropic environment, the acoustic pressure in a point of the acoustic field is given by [2]

$$p = \rho_0 \omega \frac{A}{r} \sin(\omega t - kr + \alpha) \tag{3}$$

where r is the radial coordinate, A is the amplitude of the spherical wave having the frequency $f = \omega / 2\pi$, which propagates from the source with the speed c, and k=ω/c is the wave number.

In the case when the perturbation propagates as cylindrical wave, the acoustic pressure in a point of the acoustic field can be expressed as [2]

$$p = A\left[J_0(kr) + iY_0(kr)\right]e^{i\omega t} \tag{4}$$

where r is the cylindrical coordinate, A is a constant, J_0 and Y_0 are Bessel and respectively Bessel-Neumann functions.

In the same way, taking into account that the vibration propagates as plane waves, the acoustic pressure in a point of the acoustic filed has the expression [2]

$$p = \rho_0 \omega A \sin(\omega t - kr + \varphi) \tag{5}$$

The propagation of spherical, cylindrical and plane waves is characterized by the variation of the acoustic pressure in a specific point of the acoustic field.

Using the instantaneous acoustic pressure, one can compute the level of acoustic pressure define by Eq.(1), the equivalent noise level expressed by Eq.(2) as well as percentage noise levels.

However, these indicators would be directly obtained using appropriate measurement equipment. Due to the fact that the acoustic field generated by the sources within the air conditioning installation serving the closed space of the swimming pool is very complex, for the above mentioned reasons, its investigation is recomandable to be experimentally performed. In this respect, measurements were performed using the Bruel & Kjaer investigator type 2250. In the meantime, the parameters which could influence the conditions of propagation of sound were separately recorded: wind speed, humidity and temperature.

In order to avoid road traffic noise, measurements were taken during the night, after 23:00 h in different points: in the nearby of the installation, at 2 m distance from the nearest residential building, at 2 m distance from the second nearest residential building, the microphone being moounted at 1.3 m high from the ground.

The main characteristics of the acoustic field were determined according to Romanian standards [7]-[9].

The first measurements were taken in order to characterize the noise source with the aim to establish a starting point for a future isolation system. The spectral composition of the noise generated by the air conditioning installation was obtained as is shown in fig.1, where one can identify 3 frequency bands exceeding 70 dB.

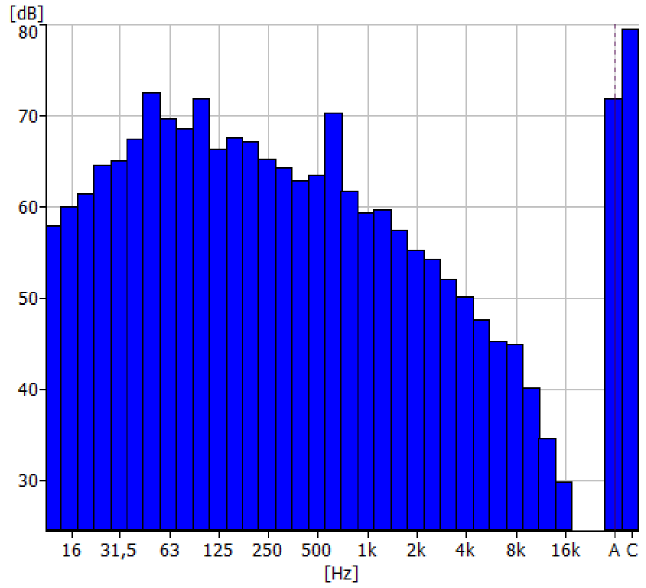

Fig. 1 *Spectral distribution of the noise source in 1/3 ocatave bands*

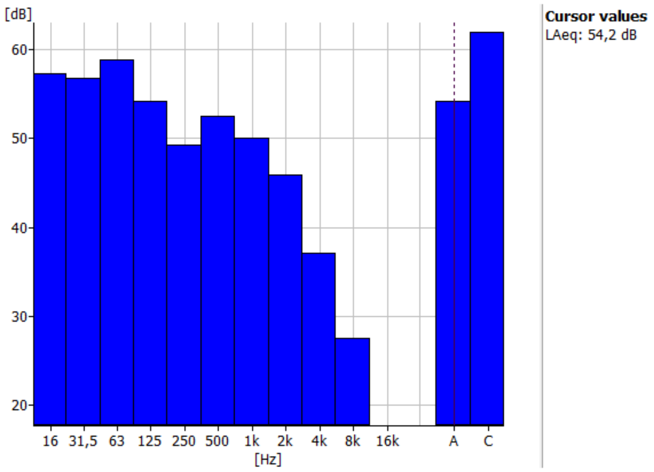

Fig. 2 *Spectral distribution of the noise in 1/1 ocatave bands for point 2 – at the edge of the footway*

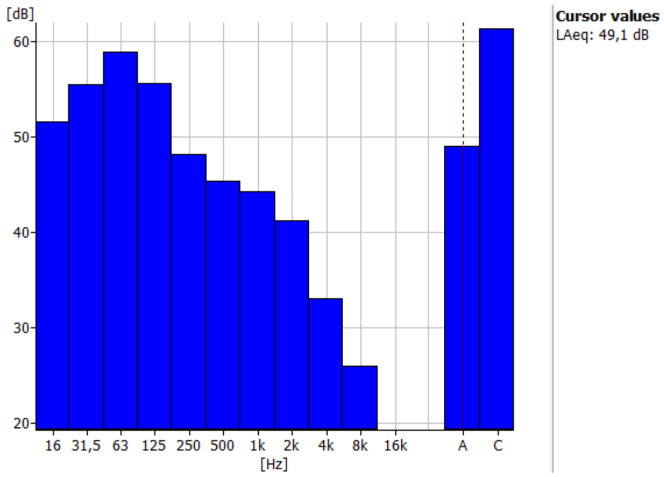

Fig. 3 *Spectral distribution of the noise in 1/1 ocatave bands for point 3 – at 2 m from the most exposed building*

The equivalent noise level obtained for the source of noise, associated to the spectral distribution from fig.1, was L_{ech}=71.9 dB(A). Informative measurements were performed in the measurement point 2 located at the edge of the footway and in the point 3 located at 2 m distance from the residential building.

The time-histrory of the noise and the spectral composition recorded in the point 2 (at the edge of the footway) and in the point 3 (in the front of the residential building) are presented in figs. 2 and 3 respectively. The equivalent noise levels in these cases were found to be L_{ech}=54.2 dB(A) in the point 2 and L_{ech}=49.1 dB(A) in the point 3.

The complete results of measurements were stored in a database concerned with the investigation of the noise generated by the air conditioning installation, developed in the frame of the Acoustics and Vibration Laboratory from the Politehnica University Timisoara, a laboratory accredited by RENAR (Romanian Accreditation Association) according to ISO 17025 for noise and vibration measurements.

5. REDUCING THE NOISE LEVEL

Following the results of measurements stored in the database and briefly presented above, one can characterize the noise sources from the air conditioning installation of the swimming pool. Since the recorded values exceed the admissible limits, one can conclude that the noise affect the life in inhabitants in the neighbourhood and some acoustic arrangements are needed for the air conditioning installation, in order to reduce the noise.

These acoustic arrangements mainly consist in an acoustic barrier mounted on 3 sides of the air conditioning installation, in the direction of residential buildings (East, North and West). Other arrangements are directed to replacing the actual wire fence with a concrete one, with better noise reducing properties, planting trees and vegetation with sound absorbtion properties, covering the existant walls with some materials with sound absorbtion properties.

In the first instance, some acoustic barriers were built on 3 sides of the air conditioning installation, oriented to the residential area. These are 3 m, 4 m and 6 m long respectively, having a high of 3.5 m. The thickness of the barrier is 0.12 m and the structure is depicted in fig.4.

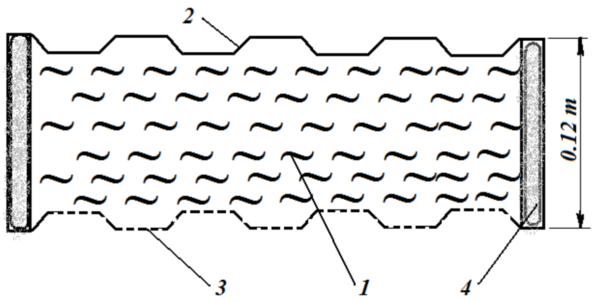

Fig. 4 *Internal structure of the acoustic barrier*

The mechanical structure is based on two metal sheets 2 and 3, one of them (denoted by 3) is perforated. Between these walls, one can find a layer of basaltic mineral wool denoted by 1 in fig.4. Finally, the structure is completed by 2 stanchions denoted by 4. This structure is designed as a modular one, so that one can construct long barriers, as log as is needed (in our case the longest had 6 m).

After installing the acoustic barrier described above, our laboratory analyzed the efficiency of this construction in reducing the noise. In this respect, new measurements were performed in the same conditions. Thus, in figs.5-8 are briefly presented the main results.

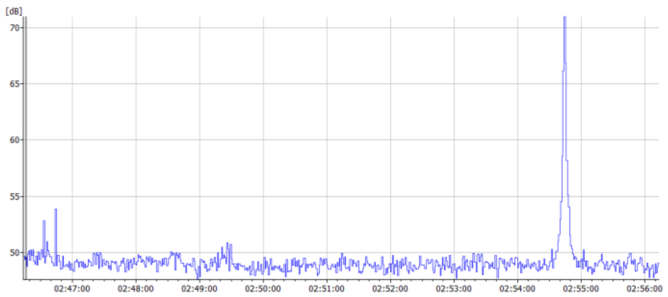

Fig. 5 *Time-history of the noise in point 2 (at the edge of the footway) after installing the acoustic barrier*

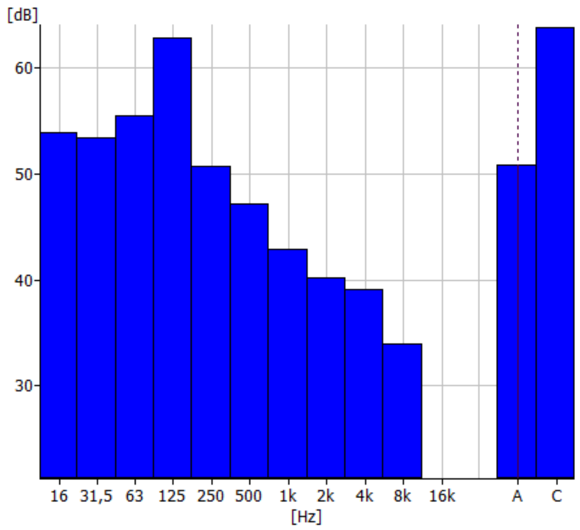

Fig. 6 *Spectral distribution of the noise in point 2 (at the edge of the footway) after installing the acoustic barrier*

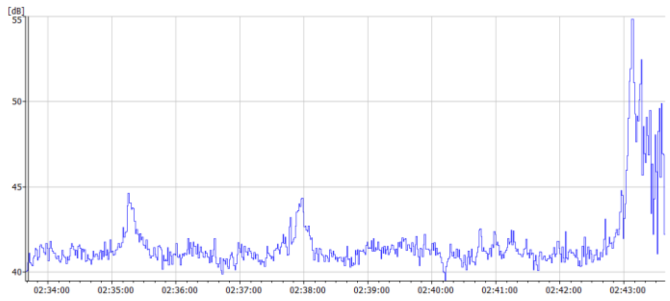

Fig. 7 *Time-history of the noise in point 3 (at 2 m distance from the building) after installing the acoustic barrier*

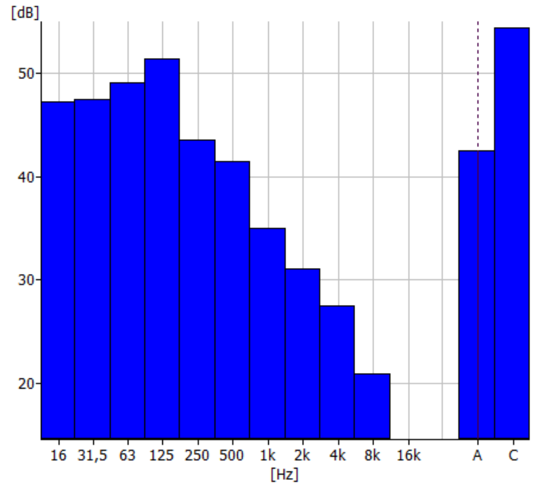

Fig. 8 *Spectral distribution of the noise in point 3 (at 2 m distance from the building) after installing the barrier*

Analyzing the results obtained by measurements after the installation of the acoustic barrier around the air conditioning installation, and comparing with the previous results, one can conclude that the noise levels were substantially reduced, being above the admissible limits.

In the case of the measurement point located at the edge of the footway, one can be seen from fig.5 that the noise level is under 50 dB(A), except the peaks recorded with the occasion of a car passing on the road. Basically, a reduction with approx 4 degree is observed in this point.

In the case of the measurement point located near the residential building, one can be observed from fig.7 a substantial reduction of the noise comparing to the initial situation, the equivalent noise level A-weighted is always below 45 dB(A), excepting the cases of cars passing on the road.

This analysis confirm the fact that the applied solution is very efficient and the environment and the life of inhabitants from the neighbourhood are not any more affected by the functioning of the air conditioning installation.

6. CONCLUSIONS

In order to establish the way in which the environment and people living in the neighbourhood are affected by the noise generated by the air conditioning installation serving the swimming pool, it is necessary to investigate the acoustic field generated by the involved sources. The values obtaine for the characteristic parameters of the noise were compared with admissible ones and after such a comparison one can establish if the residential area is affected or not and efficient method for noise reduction can be proposed. This is the case in our investigation, where after the implementation of the measures for noise reduction, a new investigation of the acoustic field was performed in order to validate their efficiency.

The proposed method for investigation and noise reduction proved to be very efficient in this particular case and could be applied also in other similar situation arising in urban areas.

REFERENCES

[1] M. Grumazescu, A. Stan, N. Wegener, V. Marinescu, *Combating the noise and vibration in technics*, Bucuresti: Ed. Tehnica, 1964.

[2] E. Ia. Iudin, *Isolation against noise*, Bucuresti: Ed. Tehnica, 1968

[3] M. Gafitanu, V. Merticaru, V. Focsa, L. Bilborosch, *Noise and vibration*, Iasi: Ed. Junimea, 1980

[4] V. Bacria, N. Herisanu, "Decreasing the noise level on a construction site", *SIMEC 2011*, pp.15-22, 2011

[5] N. Herisanu, V. Bacria, "Investigation of the acoustic field on the platform of a concrete preparation station", 4-th Int. Conf. "Noise and Vibration", Nis, Serbia, 17-19 Oct. 2012, pp.221-224

[6] Gh. Duta, "Lowering the noise in ventilation installations", *Instalatorul*, vol.7, pp.20-23, 2001

[7] *** STAS 10009-88, "Acoustics in constructions. Urban Acoustics. Admissible limits of noise levels"

[8] STAS 6161/3-82, "Acoustics in constructions. Determination of noise level in urban areas. Method of determination",

[9] STAS 7150-77, "Acoustics in industry. Methods for measuring the noise in industry"

[10] *** Larmbekamfung in Wien. Enwicklung Stand. Tendenten Magistratsabteilung Umweltschutz

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AN EXAMPLE OF NOISE ABATEMENT MEASURES FOR RAILWAY LINE

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Abstract - *Reconstruction and modernization of railway section from Beograd Centar to Stara Pazova with total length of 34.7 kilometers, provides upgrading of existing railway line in double-track line with the designed speed up to 200 km/h. The paper discusses the current state of the environment in terms of noise, and possible impacts during construction and exploitation of the railway line. Procedure for developing a simulation model of railway traffic, noise mapping, annoyance analysis, design procedure for optimizing of noise barriers and a proposal of noise abatement measures are presented. The calculation of noise propagation, annoyance analysis and optimization of noise barriers were made using simulation modeling software package "Predictor-LimA Software Suite - Type 7810". For 477 houses which were exposed to excessive noise levels during the night according to the Serbian regulations we had planned protection measures. As a primary protection measure for 410 houses eleven noise barriers with total length of 7,070 m and surface of 24,251 m^2 are planed. Rest of the houses are protected using other measurements such as, installing soundproof windows and doors or including in noise monitoring program.*

1. INTRODUCTION

This paper deals with protection measures against the noise during reconstruction, modernization, construction and exploatation of the Beograd Centar - Stara Pazova section, which is a part of the following main railway lines: E 70 Beograd – Stara Pazova - Šid - State border - (Tovarnik) and E 85 (Beograd) – Stara Pazova – Novi Sad – Subotica - State border - (Kelebia). The Beograd Centar - Stara Pazova section is 34.7 km long in total.

Beograd Centar - Stara Pazova railway line (section) is exisisting source of noise which will change existing noise levels and disruption of the population living along the tracks after modernisation. In order to determine and minimize negative effects on the population it is necessary to calculate the future noise levels through noise mapping. Also, it is necessary to do annoyance analysis and propose abatement measures in accordance with obtained results.

In the Beograd Centar - Stara Pazova railway corridor, which is 600 m wide, has been identified 9,109 buildings, out of them 5,385 are residential and noise sensitive.

The calculation of noise propagation, annoyance analysis and optimization of noise barriers were made using simulation modeling software package "Predictor-LimA Software Suite - Type 7810".

2. PROJECT DESCRIPTION

The existing double-track line from Beograd Centar to Batajnica will continue as it is but overhaul and modernization works will be undertaken. Only passenger traffic was planned on this section; freight traffic will be possible in exceptional circumstances.

Passenger and freight traffic will be merged and split in Batajnica station within the group of tracks belonging to Belgrade Railway Junction. In addition, Batajnica will be the terminal station for urban-suburban railway traffic in the City of Belgrade.

The existing right track on the Batajnica - Nova Pazova section will be moved to ensure 4.50 m distance between tracks (with 4.00 m wide formation) planned for passenger traffic. Additional two (freight) tracks spaced at 4.00 m and 6.40 m away from the existing left track are planned after Batajnica station exit. The right freight track above double-track line for passenger traffic shall be grade-separated. After grade separation (crossing) of the right freight track over the existing two tracks, both freight tracks will run at the distance of 6.40 m from the tracks for passenger traffic.

A bed for four tracks was constructed on the Nova Pazova - Stara Pazova section; two tracks are already laid along track bed edges. The design envisages laying of two central tracks. All four tracks are designed for mixed traffic. Central tracks will be used by passenger and freight trains not planned to stop in Stara Pazova station but to run between Belgrade and Novi Sad. Peripheral tracks will be used by passenger trains planned to stop in Stara Pazova station and by trains running between Belgrade and Šid.

Planned design speeds on the Beograd Centar - Stara Pazova section:

- 100 km/h - from Beograd Centar station to Novi Beograd station and on the bridge over the Sava River,
- 120 km/h - from Novi Beograd station to Batajnica station through Bežanijska kosa tunnel;
- 200 km/h - from Batajnica station to Nova Pazova station on the tracks planned for passenger traffic;
- 120 km/h - from Batajnica station to Nova Pazova station on the tracks planned for freight traffic;

- 200 km/h - from station Nova Pazova to Stara Pazova station on the central tracks (inner two tracks), and
- 160 km/h - from station Nova Pazova to Stara Pazova station on the outer tracks.

In order to make the railway line capable for the design speeds, the following activities on subsections are proposed for modernization of Beograd Centar - Stara Pazova railway section depending on the current limitations and planned traffic organization:

I. Beograd Centar - Batajnica subsection: reconstruction of existing railway line and modernization of railway devices,
II. Batajnica – Stara Pazova subsection: construction of two new tracks, reconstruction of existing tracks and modernization of equipment and devices,
III. Beograd Centar - Stara Pazova section: reconstruction of station facilities in accordance with current and new technology tasks in the stations.

According to the design, the following stations will be retained: Beograd Centre, Novi Beograd, Zemun, Zemunsko Polje, Batajnica, Nova Pazova and Stara Pazova. The current Tošin Bunar stopping place will be relocated towards Novi Beograd stations and two new stopping places, Altina and Kamendin will be open to traffic.

UIC GC loading gauge, which enables all modes of combined transport, was adopted. Cross section was designed in compliance with the Rulebook on design, reconstruction and construction of specific elements of railway infrastructure for particular main railway lines (Official Gazette of the RS, No. 100 of 19 October 2012). Safety wire fence will be placed along the entire section.

3. CALCULATION AND NOISE MAPPING

For noise calculation and noise mapping it is necessary to use a structural approach in a problem solving. To ensure the reliable output data, it is very important to well-observe the problem, define the model and all necessary input data.

During data collection, it is necessary to carry out their analysis in terms of completeness, accuracy, consistency and compliance with the requirements of the computational model.

The process of noise calculating and noise mapping produced by rail traffic can be divided into the four main phases:

1. Input data collection and preparation,
2. Calculation of noise indicators produced by rail traffic,
3. Presentation and analysis of results, and
4. Planning of noise abatement measures which will reduce negative impacts on the environment.

3.1. Collection and preparation of input data

The input data, as well as the calculation, must be done on the basis of the chosen method`s technical instructions. Regardless of the method, it is necessary to ensure that the input data are describing the situation for the calendar year prior to the calculation. In each case the data should not be older than three years.

It is essential to provide following data: about the terrain topography, soil types in terms of sound absorption, about influence of obstacles to the propagation of sound, about the track alignment including formation width, technical characteristics of railway line, railway transport data, acoustic zones through which a new railway line passes and meteorology data.

All necessary data are obtained from Preliminary design of the Modernization of the Beograd – Subotica – State border (Kelebia) railway line, Beograd – Stara Pazova section and accompanying environmental impact assessment study [1].

3.2. Calculation of noise indicators

Noise indicators were calculated and mapped with the aid of „Predictor-LimA Software Suite - Type 7810“ software package, produced by Brüel & Kjær. The method adopted for noise calculation from railway traffic is the German method „SCHALL 03 - Richtlinie zur Berechnung der Schallimmissionen von Schienenwegen“. It complies with 2002/49/EC Directive and Ordnance on noise indicators, limits, methods for their evaluation and harmful effects in the environment (Off. Gazette RS, No.75/10) as it gives the results comparable with the recommended calculation methods.

Noise indicators were calculated in a network of points 10x10 m, 2.25 m above the ground level. Measuring points needed to determine noise level were positioned on face walls of houses and of other units 0.5 m in front of them. Noise indicators were calculated using the first degree of reflection, except at face wall measuring points where reflection from the observed building was not considered. „SCHALL 03" method allows bonus of -5 dB in the calculation of noise indicators on the railways.

Maximum height of a noise barriers is limited to 5 m on the ground and to 2 m on bridge structures.

Acoustic simulations and noise indicator calculation were done using Predictor-LimA Software Suite - Type 7810 software package and maximum dynamic error of 0.5 dB(A).

In the open air noise indicator limit values may be 65 dB(A) day and evening and 55 dB(A) night. Day means time interval 6^{00} to 18^{00}, evening 18^{00} to 22^{00} and night 22^{00} to 6^{00}.

A corridor 300 m to the left and to the right from the centre line of the Beograd Centar- Novi Beograd, Beograd - Sid - state border (Tovarnik) railway line and (Beograd) - Stara Pazova - Indjija - Subotica - state border (Kelebia) railway line, precisely the section from Beograd Centre to Stara Pazova. The same corridor was used to compare existing with prospective conditions.

Impact of noise level in the surrounding can be subdivided in two segments. The first segment includes noise due to construction of new tracks and overhaul of the existing one and the other noise from train running. These impacts overlap.

3.3. Noise during construction works

Noise level during construction of a new track and overhaul of the existing one will before all depend on the site organization, number and kinds of construction equipment, their positions and distances from houses in the impact zone. As in this design stage, organization and technology of works on site are not defined, the impact of noise in the surrounding was neither modeled nor analyzed. In any case, noisy works shall be carried out at normal work hours wherever possible, with most silenced available machines, wherever suitable and cost effective, use temporary noise suppression barriers, educate workers on site on noise issues, station most noisy machines as far as possible from houses, organize delivery and haulage of material during work hours, inform the population concerned about noisy works pending etc. During those works, periodical measurement of noise shall be performed to determine whether the generated noise level exceeds permitted limit values.

3.4. Noise from railway line in operation

A study of data on prospective railway traffic volume, new line characteristics as well as 3D terrain model noise level were calculated. Data for modelling and acoustic calculations were taken from the Preliminary design for reconstruction, modernization and construction of double track railway line Beograd - Novi Sad - Subotica - Hungarian border, the section: Beograd Centar - Stara Pazova.

Solely passenger trains will run on the section Beograd Centar - Batajnica. Further on, the line will be used for mixed traffic. Along the section from Batajnica to Nova Pazova station, tracks will be divided into passenger and freight groups while from the station of Nova Pazova on tracks will be divided by directions (from/to Novi Sad and from/to Sid).

The line is designed for maximum speed of 200 km/h. Because of constraints maximum speed from Beograd Centre to Novi Beograd will be 100 km/h, from Novi Beograd to Batajnica station 120 km/h and from the station of Batajnica further on 200 km/h.

Depending on operational technology international passenger trains to Novi Sad will run at maximum speed of 200 km/h, international passenger trains to Sid at 120 km/h maximum. Maximum speed of passenger trains in regional traffic to Novi Sad will be 160 km/h, of trains to Sid 120 km/h. Commuter trains that end their journey in Batajnica station will run at maximum speed of 100 km/h. In Novi Sad direction there will be 9 pairs of international passenger trains and 31 pairs of regional trains while in Sid direction there will be 8 pairs of international and 15 pairs of regional trains. Commuters will have 53 passenger train pairs.

Maximum permitted speed of goods trains is 100 km/h in international transport and 80 km/h in domestic transport. It was assumed in the calculation that pursuing current tendencies in international transport in Europe all goods cars shall be fitted either with disc brakes or with composite brake shoes. In domestic traffic it is assumed that every second goods car shall be fitted either with disc brakes or with composite brake shoes. In the direction to Novi Sad international cargo will be hauled with 20 pairs of trains and in domestic transport with 5 train pairs. It is assumed that average net train mass in domestic transport will be 500 tons in international transport 900 tons with the coefficient of empty wagon running of 1.4.

In the calculation of noise indicators and further analysis, solely was the noise generated by railway traffic on the Beograd Centar - Stara Pazova section considered

Example of graphic presentation of noise level at night without noise barriers is given on Figure 1.

In order to assess potential noise impact on inhabitants of prospective railway traffic on the overhauled and newly laid tracks on the Beograd Centar - Stara Pazova section, measuring points were positioned in the middle of face walls of buildings exposed to noise (persons living or working in them) that are in the corridor area. Noise indicators were calculated for days, evenings, nights, day-evening-night for a total 56,700 calculation points. The number of measurement points by number of floors is shown in Table 1.

Table 1 *Number of buildings and measurement points by floors*

Floors	Number of buildings	Number of calculation points
P+0	2.240	29.002
P+1	2.281	16.509
P+2	666	4.981
P+3	135	1.523
P+4	27	742
P+5	6	578
P+6	2	534
P+7	6	472
P+8	1	397
P+9	6	369
P+10	--	275
P+11	5	275
P+12	--	150
P+13	1	150
P+14	--	146
P+15	1	146
P+16	--	137
P+17	4	137
P+18	--	59
P+19	--	59
P+20	4	59
Total:	5.385	56.700

* P = ground floor

Acoustic zoning was not done in the project area. In order to analyse impact of noise on population and if necessary to plan protective measures it was necessary to assume, pursuant to Ordinance on noise indicators, limits, methods for their evaluation and harmful effects in the environment (Off. Gazette RS, No.75/10) that the corridor considered here belongs to zone 5 (City centre, crafts, commercial, administrative-government zone with apartments, zone along motorways, main and city avenues). Limit values for noise indicator for zone 5 during day are Lday = 65 dB(A), evening Levening = 65 dB(A) and night Lnight = 55 dB(A).

Noise indicator at night was taken as ultimate parametre for further analysis. This was based on the fact that excessive values on all measurement points occurred at night namely there were no measurement points where excessive values occurred only during day or evening.

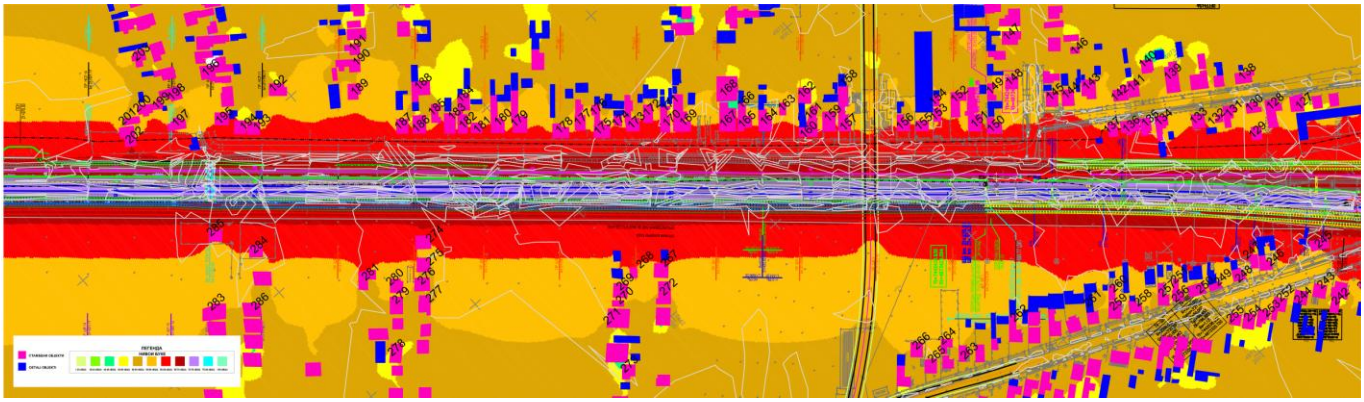

Fig. 1 *Example of graphic presentation of noise level at night without noise barriers*

It was calculated that on face walls on 477 housing units the noise level exceeded the permitted values at night (Lnight > 55 dB(A) which accounts for 8.9% of the total number of noise sensitive buildings. Excessive values were found on 1.701 face walls and/or calculation points. The number of calculation points where excessive values appeared is shown in Table 2.

Table 2 Number of measuring points with Lnight > 55 dB(A) against number of floors

Floors	Number of measurement points
P+0	855
P+1	632
P+2	159
P+3	37
P+4	3
P+5	3
P+6	6
P+7	6
Total:	1.701

* P = ground floor

3.5. Noise abatement measures

To reduce negative railway-borne noise impact on the environment and on population, measures of protection shall be planned and implemented. This shall be done wherever noise levels exceed legally permitted values.

Protection measures shall be planned and implemented only for buildings in which people live and dwell, namely buildings sensitive to noise such as kinder gardens, elementary and secondary schools, university buildings, health centres and hospitals. Attention shall be paid to working hours in those noise – sensitive buildings.

Basic noise barriers are therefore designed. In the relevant analysis and option optimisation, attention was paid not only to the existing buildings but also to planned land uses and development plans.

No noise suppression measures are planned for buildings on railway land. There are a total of 22 buildings on the railway ground for which protection measures are not planned.

They are not planned either in the spatial and urban plans for the settlements of Altina and Kamendin for 18 buildings which are located on the route of a future city avenue. The spatial and urban development plans for the Plavi Horizonti settlement include 11 buildings which are situated in the railway right-of-way in which no building construction is allowed. For this reason no protective measures are planned. Therefore, the spatial and urban plans include a total of 29 buildings for which no protective measures are planned.

The noise suppression plan does not include 51 buildings namely 10.7% of the total number of affected buildings. Noise suppression measures are planned for 426 buildings, namely 89.3% of the total number of affected buildings.

For the purpose of protection of the affected housing and other sensitive buildings, an outcome of an analysis and optimization attempt was to build 11 noise barriers, in the total length of 7,070 m and 24,251 m^2 in area. Basic details on planned noise barriers, heights of their parts, length and area are given in Table 3. Example graphic presentation of noise levels for night periods with noise barriers in place is shown on Figure 2.

Table 3 Main data about noise barriers

Barrier No.	Height	Length	Area
	[m]	[m]	[m^2]
1	2.0 - 4.0	366	863
2	2.0 - 4.0	241	772
3	2.0 - 4.0	360	1,050
4	2.0 - 3.5	280	640
5	2.0 - 4.0	400	1,080
6	2.0 - 4.0	1,344	3,858
7	2.0 - 5.0	1,204	5440
8	2.0 - 4.0	883	3112
9	3.5 - 5.0	504	2390
10	2.0 - 4.0	1,024	3,150
11	2.0 - 5.0	464	1,896
Total		**7,070**	**24,251**

Noise barriers on the ground shall be made of absorption materials while barriers on bridges shall be made of transparent materials. In order to provide better comfort to passengers, all noise barriers or at least the long ones at the level of passenger car windows should be transparent.

At “Senjak” tunnel exit towards Novi Beograd the tunnel walls should be lined with absorbing panels in the length of about 12 m, 3.5 m high as additional noise suppression measure.

All acoustic panels shall have CE marking in accordance with SRPS EN 14388 standard. In compliance with the listed standards, all acoustic panels that will be used for noise barriers shall have sound absorption of minimum 12 dB (class A4 in accordance with SRPS EN 16272-1) and soundproofing of minimum 30 dB (class B3 in accordance with SRPS EN 16272-2).

All members of noise barrier shall be grounded. The acoustic panels shall have service life of minimum 20 years without major changes in their acoustic and non-acoustic performances.

The acoustic panels and/or complete noise barrier shall be suitable for installation next to railway lines with max. permitted speeds of 200 km/h.

Noise barriers longer than 300 m shall be provided with accessible emergency door. Adequate access ways shall be provided to all emergency doors.

Standard cross sections of railway line provided with two parallel noise barrier in Figure 3.

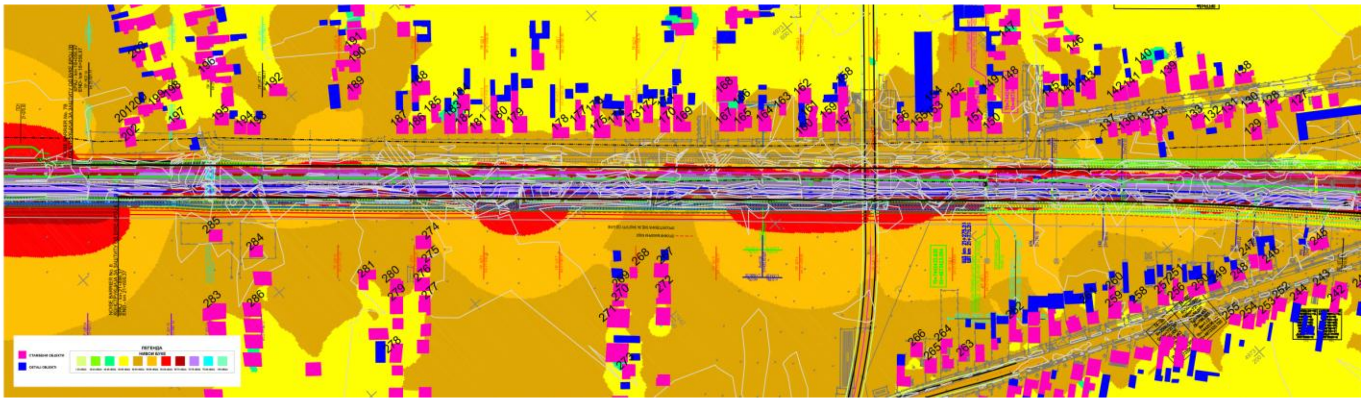

Fig. 2 *Example of graphic presentation of noise level at night with noise barriers*

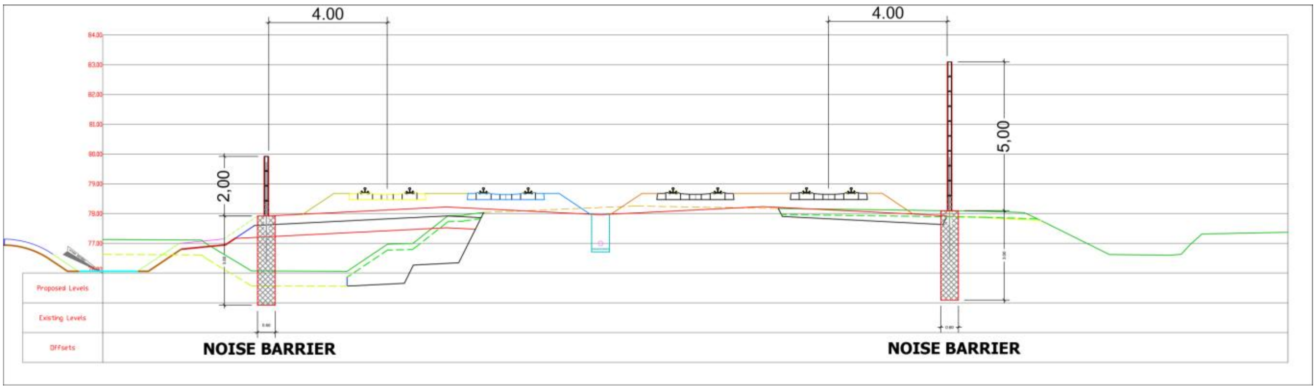

Fig. 3 Standard cross sections of railway line with noise barriers

The design of noise barriers shall comply with provisions of the Rulebook on design, reconstruction and construction of specific elements of railway infrastructure for particular main railway lines (Official Gazette of the RS, No. 100 of 19 October 2012), other relevant national and foreign legislation and positive experience of foreign and local practice.

Position of noise barriers by districts, number of buildings they protect and their efficiency are shown in Table 4.

For residential buildings and other sensitive buildings which protection by noise barriers is not economical or technically possible and for buildings where exceeding noise level occurs even after installation of noise barriers, some other protection measures were planned such as replacement of doors and windows with better sound insulated ones. Decision on the type of sound insulation (sealing glass) will be made separately for each case, with a note that small sound insulation will not resolve the above-mentioned problems while big sound insulation is not economic due to very high prices. For each building protected by replacement of doors and windows with those having better sound insulation, closed fresh air supply system should be provided as well. In addition to replacement of doors and windows on the buildings, the facades should be provided with adequate soundproofing. Disadvantage of such approach is that noise level outside the building and/or in the yards are not reduced.

Condition of tracks and rolling stock has the biggest impact on railway noise emission and therefore the regular maintenance is planned as one of the most important noise suppression measures. Planned noise barriers will fulfil their main function only if tracks and rolling stock are in good state and regularly maintained.

Table 4 Position of noise barriers by districts, number of buildings they protect and their efficiency

Barrier No.	District	Number of buildings they protect	Efficiency [dB]
1	Senjak (Beograd)	9	16.3
2	Senjak (Beograd)	10	16.5
3	Zemun (Beograd)	10	10.3
4	Zemun (Beograd)	4	4.8
5	Batajnica (Beograd)	22	8.4
6	Batajnica (Beograd)	83	10.6
7	Batajnica (Beograd)	94	14.8
8	Nova Pazova (Nova Pazova)	57	9.4
9	Nova Pazova (Nova Pazova)	46	11.0
10	Stara Pazova (Stara Pazova)	56	16.3
11	Stara Pazova (Stara Pazova)	19	8.5

3.6. Noise monitoring

Upon the start of the railway line exploitation, noise monitoring shall be undertaken in order to determine actual noise conditions together with periodical control measurements for further noise monitoring.

Noise monitoring during operation of the railway line operation shall serve to predict and implement adequate measures. Optimum technical measures for noise suppression can be planned only after a series of noise measurements with the railway section in operation and over a long period of time, as noise level in one moment and in one point cannot be representative of noise at that place or in a community. Besides, any optimum protective measures may be exposed to noise from other sources and may depend on the very structure of buildings in the zone of influence. Partial concepts may be counterproductive for the environment (reflection, superposition and other).

Noise monitoring shall be undertaken in zones with housing units and other sensitive buildings to be provided with noise suppression barriers. The results shall serve to confirm efficiency of erected noise barrier.

Environmental monitoring from the aspect of noise is envisaged at places where excesses are expected, before all in housing areas in the proximity of the railway line.

Points of measurement shall be representative for the project area and their number may be increased in case of justified complaints of the local population. If measurements show further excessive noise levels from the ones determined as well as some new overrun values, the Investor, namely the institution in charge shall comply.

4. CONCLUSION

Modernized and upgaded line from Beograd Centar to Stara Pazova, as old one will significantly contribute in increasing existing noise levels in areas along the railway. Noise from railway will threatened 477 residential buildings which represents 8.9% of total number.

Abatement measures are planned as follows:

- Noise barriers for protection of 410 object;
- New joinery with better sound insulation for protection of 16 objects.

No noise suppression measures are planned for buildings on railway land, for buildings where the spatial and urban plans define the route of a future city avenue and for buildings which are situated in the railway right-of-way in which no building construction is allowed. There are a total of 51 buildings for which protection measures are not planned.

The values of noise levels are given on the basis of calculation. After the railway will be put into operation it is necessary to carry out the noise monitoring. If the measured values of noise levels will exceed permitted levels, appropriate additional noise abatement measures will be applied.

ACKNOWLEDGEMENT

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REFERENCES

[1] *Preliminary design of Modernization of the Beograd – Subotica – State border (Kelebia) railway line, Beograd – Stara Pazova section, in Beograd, New Beograd, Zemun, Zemunsko Polje, Batajnica, Nova Pazova, Stara Pazova on the building plots according to the list attached to documentation*, Saobraćajni institut CIP, Beograd, 2016 (In work).

[2] M. Praščević, A, Gajicki, D. Mihajlov, N. Živković, Lj. Živković, „*Application of the prediction model "SCHALL 03" for railway noise calculation in Serbia"*, Applied Mechanics and Materials, Vol. 430, pp 237-243, 2013.

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MEASUREMENTS OF NOISE OF DIESEL MOTOR TRAIN OF SERIES ŽS 711

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Abstract - *Serbian Railways recently introduced several new types of rail vehicles in operation. In 2012, multiple diesel unit of class 711, manufactured by Russian company "Metrovagonmash" was purchased by the Serbian railroad operator. These train units are expected to replace old diesel rail-buses that operate on sections without electric power, and, therefore, to become the dominant type of rail vehicles on Serbian local railway lines. This paper presents results of measurements of the equivalent level and one-third octave spectrum of noise emitted by a stationary unit of class 711 and a class 711 train moving at constant speed. All measurements are performed according to the ISO 3095:2013 standard.*

1. INTRODUCTION

Environmental noise is frequently described as undesirable sound caused by transport, industrial or recreational activities [1]. According to the World Health Organization, noise can significantly affects human health and productivity, causing stress, sleep disturbance, cardiovascular problems, etc., as it has been shown in many studies during past decades [2-4].

The major sources of noise pollution in the living environment are transportation systems [5]. European authorities are trying to increase the transport of passengers and goods by rail transport systems due to economic aspects, but also in order to reduce GHG emissions. Increase of the volume of the rail transport will lead to increase of the railway noise pollution.

2. SOURCE OF NOISE ON RAIL VEHICLES

Rail vehicles are complex mechanical systems, with multiple noise sources that emit sound waves due to [6]:

- interaction between contact surfaces in wagon assemblies,
- interaction between the outer surfaces of locomotives and wagons with air during motion,
- interaction between the track and the substrate,
- interaction between wheels and breaks,
- running of basic and additional equipment in locomotives and wagons.

Based on type of interaction, noise of rail vehicles may be classified in following categories [7]:

- rolling noise,
- curve squeal,
- aerodynamic noise,
- bridge noise,
- ground noise and vibration,
- internal noise and vibration,
- traction noise.

The wheel – rail contact represents dominant noise source of rail vehicles for speeds within the range between 40 km/h and 140 km/h. Rolling noise, which is caused by the interaction between the wheel and the track, increases with train speed v at a rate of about $30\log(v)$ [7]. Traction motors, engines and railway vehicle equipment are main sources of traction noise, which dominates at lower speeds. The aerodynamic noise, caused by air motion around vehicle and pantograph system, is dominant at higher speeds (over 140 km/h). Further, rail vehicles in the curves produce a squeal noise due to sliding of the wheel over the top of the rails. The various infrastructure objects, such as bridges and tunes, also have great influence to railway noise generation.

Table 1 *Frequency range for different types of railway noise*

Noise type	Frequency range [Hz]
Rolling	250 - 5000
Flat spots	50 – 250 (function of speed)
Ground borne vibrations	4 – 80
Engine	50 - 250
Top of rail squeal	1000 - 5000
Flanging noise	5000 - 10000

Frequency ranges for different types of railway noise are shown in Table 1.

3. MEASUREMENTS

In March 2012, multiple diesel unit of class 711 began commercial service from Belgrade to Vršac, Serbia. These new units are planned to replace old railway coaches that run on the sections without electric power by the end of 2016.

Fig. 1 *Class 711 diesel multiple unit train*

The characteristics of the ZS 711 multiple train units are given in Table 2.

Table 2: *ZS 711 characteristics*

Weight (two units)	109 t
Length(two units)	45 m
Width	3.140 m
Engine type (per unit)	Diesel
Power output	2 x 250 kW
Maximal speed	120 km/h
Operating speed	100 km/h
Maximum number of passengers	246

Considering that the Class 711 diesel trains are relatively new and information about their noise emission is not available, this paper presents the results of the measurements of the stationary unit noise, as well as the noise emitted by the train unit running at a constant speed. The measurements were performed according to the ISO 3095:2013 standard [8].

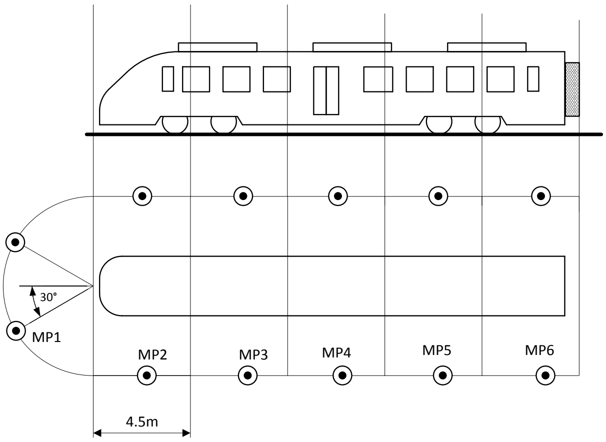

Fig. 2 *Measurement points for stationary speed test*

Measurement positions were located at distance 7.5 m from the track centerline, at a height of 1.2 m above the rail upper surface. The area within radius three times larger than the distance between the measurement point and the track centerline was free of large reflecting objects. Further, the area between the track and the microphone position was free of natural and artificial obstacles to sound propagation, sound absorbing matters and reflective coverings.

3.1 Stationary test

The noise emitted by a stationary unit was measured using a Bruel&Kjaer hand-held analyzer type 2270. The car was divided into areas with identical horizontal length of 4.5 m, and five measurement positions were located at middle of the corresponding areas. An additional measurement position was located at 30° from the track centerline, on a circle having a radius of 7.5 m and center in the midpoint of the unit end, as shown in the Figure 2. The measurements were performed only on one side of the unit, since both sides of the unit are acoustically identical. Ground surface was in level with the top of the rail surface.

3.2 Constant speed test

Bruel&Kjaer PULSE analyzer was used for measurement of the noise produced by railway vehicle that ran at a steady speed. Free-field microphones and a suitable microphone windscreen were used. The vehicle moved at a speed of 60 km/h along the track whose curvature has the radius larger than 1000 m.

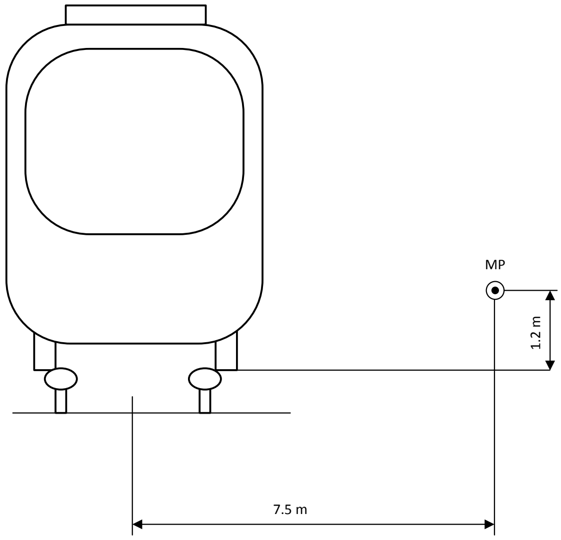

Fig. 3 *Measurement point for constant speed test*

The location of the measurement point relative to the rail track is shown in Figure 3. The ground surface level was up to 1.5 m below the top of the rail. The acoustic rail roughness has not been observed. The state of the rail running surface is shown in Figure 4.

Fig. 4 *Track condition on the measuring site*

4. RESULTS

Equivalent A-weighted noise level and one-third octave spectrum of the stationary unit were measured during 20 s time interval. The obtained noise levels at each measurement positions are given in Table 1.

According to ISO 3095 standard, noise emission level of the unit shall be calculated by energy averaging of $L^{i}_{pAeq,T}$

measured noise levels at all positions i according to the following equation:

$$\left\langle L_{pAeq,T} \right\rangle_{unit} = 10\log\left(\sum_{i=1}^{n} \frac{l_i}{l_{tot}} 10^{L^i_{pAeq,T}/10}\right), \quad (1)$$

where n is the number of measurement positions, l_i length of the wagon part where measurement position MPi is located, while l_{tot} represents the sum of lengths of all n wagon parts. Calculated noise emission level of the stationary unit of class 711 has value 72.63 dBA, and, therefore, satisfies the acoustic requirements of the Technical Specifications for Interoperability (TSI) [9], which was introduced for the purpose of harmonization of European rail traffic and reduction of rail traffic noise. TSI defines noise level of 73 dBA as the limiting value for the stationary noise of the diesel multiple unit.

Table 3 *Stationary unit noise levels*

	MP1	MP2	MP3	MP4	MP5	MP6
LAeq	66.93	70.93	74.59	75.36	75.09	72.73
31.5 Hz	36.28	38.98	40.26	37.48	34.63	29.23
40 Hz	42.38	43.72	44.97	42.37	45.96	41.78
50 Hz	28.73	30.57	31.05	29.43	29.88	30.11
63 Hz	28.75	30.77	35.2	35.39	35.75	35.67
80 Hz	32.78	34.74	39.16	39.79	41.85	39.78
100 Hz	35.5	41.33	41.99	41.19	42.56	42.55
125 Hz	34.76	36.01	39.88	42.5	43.45	41.98
160 Hz	31.21	32.81	35.54	40.55	37.6	35.27
200 Hz	36.48	41.32	45.95	45.73	46.31	40.96
250 Hz	44.35	50.2	52.67	49.87	57.48	51.42
315 Hz	42.96	50.2	50.46	52.12	52.51	53.22
400 Hz	45.75	49.1	52.24	52	51.01	53.8
500 Hz	53.83	56.32	58.69	61.65	58.17	51.58
630 Hz	55.66	60.67	66.59	66.51	66.35	62.89
800 Hz	55.5	61.59	69.2	66.76	66.87	65.14
1 kHz	59.94	61.82	65.47	68.85	67.07	64.27
1.25 kHz	56.68	63.08	64.45	66.36	66.68	62.87
1.6 kHz	57.32	61.08	61.45	63.21	63.42	62.98
2 kHz	59.39	62.54	64.43	65.97	66.61	64.51
2.5 kHz	55.09	58.39	60.48	61.97	62.24	60.03
3.15 kHz	52.23	55.28	57.92	59.44	58.75	57.09
4 kHz	55.24	59.52	61.15	61.87	62.37	60.58
5 kHz	49.38	52.01	54.31	55.75	55.75	53.8
6.3 kHz	47.2	50.3	52.51	53.75	53.67	51.63
8 kHz	44.83	47.93	51.04	51.64	51.33	49.29

Figure 5 shows one-third octave noise spectrum at measurement position with the highest measured equivalent noise level (MP4). Noise levels of the unit were more than 10 dB above the background noise level, which was measured during a time interval of 20 s.

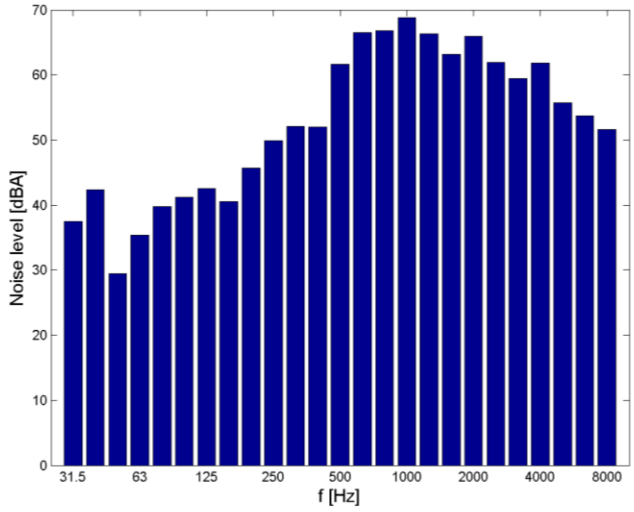

Fig. 5 *One-third octave noise spectrum at MP4*

For the constant speed test, equivalent A-weighted noise level and one-third octave noise spectrum were measured during time interval of passage of the whole unit by the measurement position. The obtained noise levels, as well as the background noise measured during 20 s time interval, are shown in Table 3.

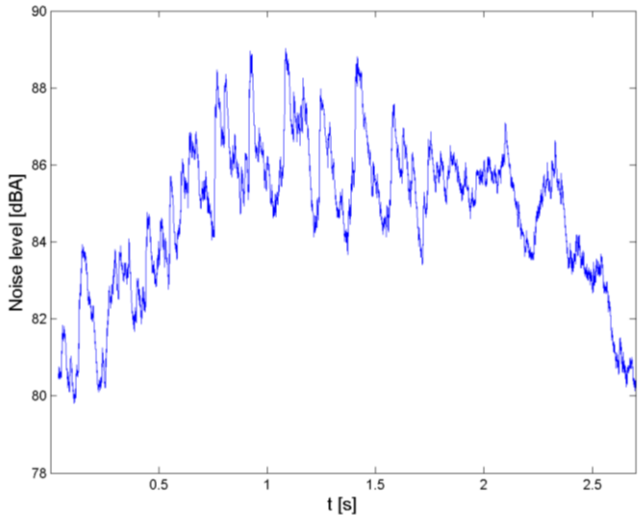

Fig. 6 *Railway vehicle pass-by noise*

Measured sound pressure level during railway vehicle pass-by is shown in Figure 2, while the Figure 3 shows corresponding one-third octave noise spectrum.

5. CONCLUSIONS

The measurements of the noise generated by multiple unit of class 711 in stationary conditions and during vehicle running have been presented in this paper. The obtained A-weighted levels of the equivalent noise are very close to the limit values requested by TSI specifications.

In the case of the stationary unit noise measurement, the highest values of the noise levels are obtained at the measurement point MP4, which is located in front of the diesel engines. The lowest noise level is measured in front of the vehicle, at measurement point MP1.

Table 4 *Vehicle pass-by noise levels*

	$L_{A,eq}$ [dB]	
	Rail noise	Background noise
f [Hz]	85.09	49.87
31.5	30.128	13.725
40	38.522	15.895
50	39.235	21.03
63	42.722	21.308
80	55.513	20.868
100	57.172	22.512
125	61.76	23.175
160	64.347	25.76
200	65.601	29.976
250	65.862	37.826
315	69.009	41.725
400	72.443	38.863
500	75.844	41.224
630	76.603	41.224
800	76.767	40.869
1000	77.33	40.064
1250	73.986	38.966
1600	71.795	37.084
2000	71.617	35.188
2500	71.055	32.599
3150	69.697	29.188
4000	68.853	26.352
5000	65.992	23.123
6300	63.848	19.483
8000	62.604	15.958

While the equivalent noise levels at three measurement points (MP4-MP6) are higher than 75 dBA, the average stationary noise of this type of vehicle satisfies the acoustic requirement of TSI specification.

Since the limit value of 85 dB, which is defined for the pass-by noise of diesel multiple unit at speed of 80 km/h, is achieved by the vehicle that runs at 60 km/h, noise emission of class 711 unit is expected to exceed the limit value recommended by TSI for the vehicle speed of 80 km/h.

Considering that rail acoustical roughness has not been taken into account, the emitted noise level of this type of vehicle may deviate significantly from presented results at different railway sections.

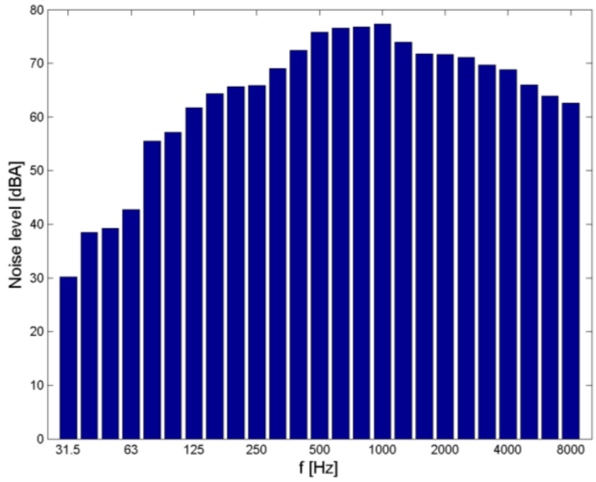

Fig. 7 *One-third octave spectrum of railway vehicle pass-by noise*

In order to fully characterize the noise emission of class 711 diesel unit, the series of pass-by noise measurements should be performed at different vehicle speeds. Further, all important track parameters and their influence on noise emission should be determined.

ACKNOWLEDGEMENT

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REFERENCES

[1] Department of Environment Heritage and Local Government, Ireland, 2006. Statutory Instrument Number 140 of 2006, Environmental Noise Regulations.

[2] Ohrstrom, E.; Rylander, R. "Sleep disturbance by road traffic noise - A laboratory study on number of noise events". Journal of Sound and Vibration, vol. 143, no. 1, pp. 93–101, 1990.

[3] Fyhri, A.; Klboe, R. "Road traffic noise, sensitivity, annoyance and self-reported health - A structural equation model exercise". Environment International, vol. 35, no. 1, pp. 91-97, 2009.

[4] Pirreera, S.; Valck, E. D.; Cluydts, R. "Nocturnal road traffic noise: A review on its assessment and consequences on sleep and health". Environment International, vol. 36, no. 5, pp. 492–498, 2010.

[5] Stansfeld S., Haines M, Brown B., Noise and health in the urban environment, Rev Environ Health. 2000 Jan-Jun;15(1-2):43-82. PMID: 10939085

[6] Predrag Petrović, Živojin Petrović, Stanislav Glumac, Railway noise resources, key environmental problem in the European union, Noise and Vibration, Niš, Serbia, 2014.

[7] Thompson, D., Railway noise and vibration – Mechanisms, modeling and means of control, Elsevier Ltd, UK, 2009.

[8] BS EN ISO 3095:2013 – Acoustics – Railwau application – Measurement of noise emmited by railbound vehicles (ISO 3095:2013)

[9] TSI Noise, 2011-04-13. Commission Decision of 4 April 2011 concerning the technical specifications of interoperability relating to the subsystem ‘rolling stock – noise’ of the trans-European conventional rail system (2011/229/EU).

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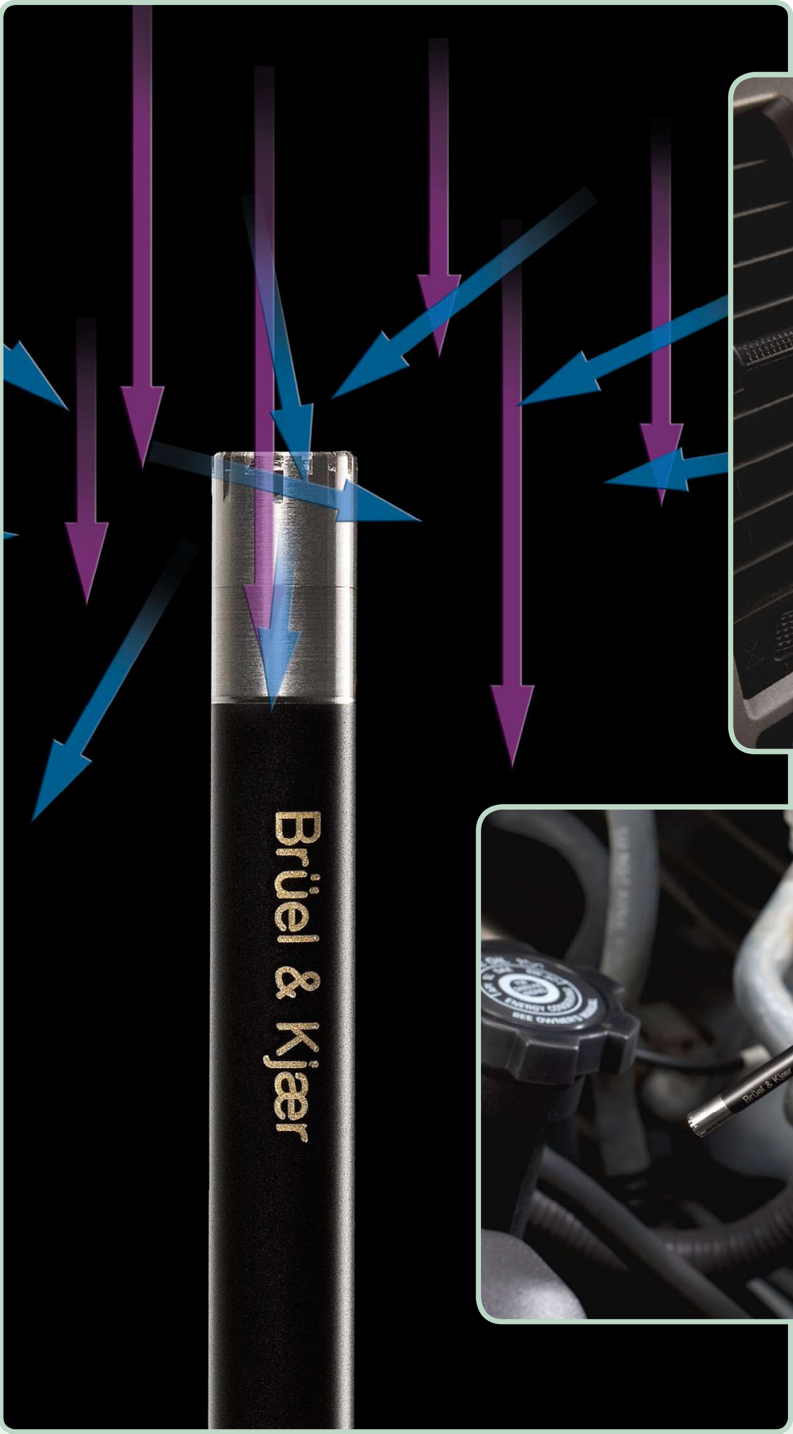

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ASSESSMENT AND EVALUATION OF NOISE AT WORKPLACE ACCORDING TO THE NEW REGULATION AND APPLICABLE STANDARDS COMPARING TO THE OLD REGULATION

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Abstract - *With the entry into force of the new Noise Regulation (Official Journal, no. 96/11 and no. 78/15) stopped the application of the old Noise Regulation (Official Journal, no. 21/92). While the new Regulation prescribed the method of determination of the noise exposure at the workplace (ISO 9612) and prescribed the noise exposure limits with respect to risk of hearing damage, there were no specified criteria left for the evaluation of noise level range underneath the risk of hearing damage, that the old Regulation divided into four categories by type of interference (interference with work activities, interference with speech communication, indirect communication jamming and interference with sound signal recognition) and for each gave the exposure limits. In practice of the workplace noise testing the requirements for the assessment of noise level ranges underneath the risk of hearing damage are common. This paper presents the guidelines for assessing and evaluating noise also for such cases, according to existing standards.*

1. INTRODUCTION

Until January 2016 the assessment and evaluation of noise at workplace were prescribed by the Regulation on occupational safety measures and norms for noise protection at workplace [1] (in further text: the Old Regulation). Since then the new regulation is into force, the Regulation on preventive measures for health and safety at work regarding noise exposure [2] (in further text: the New Regulation), that presents the national implementation of the Directive 2003/10/EC [4]. Regarding the assessment of noise at workplace, New Regulation prescribes ISO 9612 method [4] as the method of determination of the noise exposure at the workplace. As for the evaluation of noise at work, while according to the Old Regulation the evaluation was made comparing the measured (rating) levels (A-weighted equivalent continuous sound pressure level, sound pressure levels in octave bands and A-weighted peak sound pressure level) to the limit values divided into five categories (interference with work activities, interference with speech communication, indirect communication jamming, interference with sound signal recognition and risk of hearing damage), the New Regulation prescribes only the noise exposure limits with respect to risk of hearing damage and the evaluation is made by comparing the three noise quantities to the limit values (A-weighted noise exposure level normalized to a nominal 8 h working day, A-weighted noise exposure level normalized to a nominal 8 h working day averaged over the 5 day working week and C-weighted peak sound pressure level).

2. INSTRUMENTATION AND MEASUREMENT METHOD

Instrumentation requirements, measurement methodology and procedure, the basic measurement quantity (A-weighted equivalent continuous sound pressure level) and measurements' positions are similar by both Noise Regulations. The new in New Noise Regulative (in a sense that that is not likely to be reduced to the procedure required by the Old Noise Regulative) is job-based measurement strategy (see 2.2.2 and 2.2.3). At the end of this chapter the summary of this comparative review is presented in Table 1.

2.1. Instrumentation

2.1.1. Sound level meters and personal sound exposure meters

According to Old Regulation instrumentation compliance to IEC 651 Type 1 [6] and IEC 804 [7] was mandatory. ISO 9612 and consequently the New Regulation demand the sound level meters to meet the requirements for IEC 61672-1:2002 [8] Class 1 or Class 2 and the personal sound exposure meters (personal noise dosimeters) to meet the requirements for IEC 61252:2002 [10], while Class 1 instrumentation is preferred.

Standards IEC 651 (that relates to the specification for sound level meters, later renamed IEC 60651) and IEC 804 (that extends IEC 60651 and describes the performance of integrating sound level meters, later renamed IEC 60804), both superseded by IEC 61672 [8, 9], differed the grades in order of accuracy Type 1 and Type 2. The most recent standard IEC 61672 differs Class 1 and Class 2. Most sound level meters that meet the requirements of IEC 60651:2001 and IEC 60804:2000 also meet the acoustic requirements of IEC 61672-1:2002, and former Type 1 and Type 2 correspond to Class 1 and Class 2.

Standard IEC 61252 relates to the specifications for personal sound exposure meters of one accuracy grade.

In addition, by the Old Regulation the level meter was to provide the one-third-octave band analysis ([1], Clause 12, sub clause 3), even though the octave-band analysis was satisfactory by the method. For the requirements for the quantities to be measured see 2.2.1.

2.1.2. Calibrator

ISO 9612 (and consequently the New Regulation) demand the requirements of IEC 60942:2003 class 1 [11] for the calibrator. Not specified by the Old Regulation, but implicitly the instrumentation of class 1 of the former IEC 60942 was to be used according to the required Type 1 of sound level meters. Also, national standard JUS N.R6.032 [12], then mandatory (national version of IEC 179 [13]), did have a requirement for such calibrator ([13], Clauses 8.8. and 9.1, sub clause 14).

2.1.3. Periodic verification

ISO 9612 recommends that the sound calibrator and the instrumentation system are verified at intervals not exceeding 2 years, unless national regulations dictate otherwise. Regulation on the types of measurement instrumentation for those that the verification is mandatory and on the time intervals of their periodic verification [14] dictates 2 year interval, and so did the former regulation, so the periodic verification of the instrumentation stays practically the same.

2.2. Measurement method

2.2.1. Noise quantities to be measured

Both the Old Regulation and the New Regulation (i.e. ISO 9612) demand the measurement of A-weighted equivalent continuous sound pressure level, $L_{p,\mathrm{A,eq}T}$ (in further text: equivalent sound pressure level, L_{pAeq}), the basic measurement quantity. In addition, by Old Noise Regulation the measurement of sound pressure levels in octave bands and A-weighted peak sound pressure level was required, if relevant (see 4.1). Unlike, the New Regulation requires the measurement of C-weighted peak sound pressure level, $L_{p,\mathrm{Cpeak}}$.

2.2.2. Measurement procedure

Measurement procedure is similar, though minutely defined by the New Regulation, i.e. ISO 9612.

By the Old Regulation, measurements are performed for all the places the worker is working during a day shift. If the noise levels vary depending on working conditions the procedure is given ([1], Annex 1) to provide an adequate assessment with a given accuracy. If the noise levels vary from working day to working day the average week value of measured (rated) noise levels should be assessed and evaluated. If the noise levels vary greatly depending on working conditions, measurements made with personal noise dosimeters during entire work task performance are preferred. Measurement procedure that comprises all these elements yet more elaborated is given in ISO 9612.

ISO 9612 gives 3 measurement strategies: task-based measurement, job-based measurement and full-day measurement. Selection of strategy depends on the activities of the enterprise, jobs of workers under consideration and significant noise events. Also, related to the selection of strategy is the instrumentation selection, e.g. for full-day measurement it is practical to use personal sound exposure meters. The selection of measurement strategy is made based on work analysis, optimizing the effort required for obtaining appropriate accuracy. An example illustrating the hierarchy of jobs and tasks ([4], Figure 1) is given in Figure 1, and an example of different noise situations during the performance of different tasks ([4], Figure 2) is given in Figure 2.

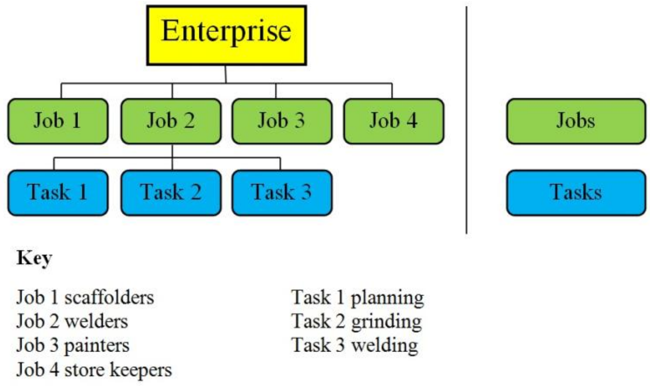

Fig. 1 *An example illustrating the hierarchy of jobs and tasks*

2.2.3. The duration of measurement

By the Old Regulation, duration of each measurement depends on the type of the noise, and shall be long enough to ensure that measured L_{pAeq} is representative of the cycle of the changes of the noise levels in time. An example of duration of measurement depending on the type of noise is shown in Figure 2. Further, if the noise is non-steady, measurements shall be made three times during the day shift and two times during the night shift. For the day shift the measurement duration shall be less than 16 h (starting at 6.00 a.m. and finishing at 10.00 p.m.) and for night shift less than 8 h (starting at 10.00 p.m. and finishing at 6.00 a.m.).

By the New Regulation, i.e. ISO 9612, the duration of measurement depends on the measurement strategy chosen.

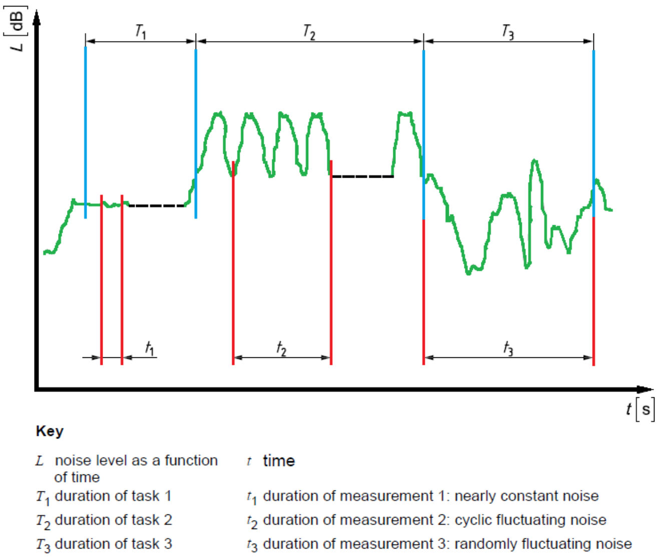

Fig. 2 *An example of three periods with different noise situations and actual duration of each measurement*

For task-based measurement duration of each measurement shall be long enough to ensure that measured L_{pAeq} is representative of the whole of the task (see Figure 2). While this is practically the same as by the Old Regulation, both examples are represented by the same figure. The duration of measurement should be at least 5 min, except if the duration of the task is shorter than 5 min or if the level is found to be constant or repeatable, then the duration shall be equal to the duration of the task, or may be reduced. For cyclic noise the measurement shall cover the duration of at least three cycles. For each task at least three measurements should be obtained due to measurement uncertainty considerations.

For job-based measurement, minimum cumulative duration of measurement to be distributed over the homogenous exposure group depends on the number of workers in the group and varies from 5 to 17 h ([4], Table 1).

For full-day measurement duration shall be over the entire working day, or over as large part of the day as possible, covering all significant periods of noise exposure. At least three full-day measurements shall be taken.

One may easily see that the essentially new approach in the measurement methodology is job-based measurement strategy, while the other two strategies can easily be seen as an elaboration of the Old Regulation's methodology.

2.2.4. Field calibration

Implicitly by the Old Regulation (not stated, but required by the instrumentation used) calibration check had to be performed at the start of the daily series of measurements. By the New Regulation (ISO 9612) the calibration check with appropriate adjustment shall be performed before each series of measurements, and in addition shall be performed after the series of measurements, and if the readings before and after differ by more than 0,5 dB the results of that series shall be discarded.

2.2.5. Microphone position

By the Old Regulation, the microphone shall be placed on the worker's position, in the height of his ears, at the distance of 0,2 m from the ear. When the worker's position is not well defined measurements shall be performed at the position that is representative of the load of the worker and without the worker's presence, in the height of 1,6 m above the ground when working standing, and 1,2 m above the ground when working seated.

By ISO 9612, when using personal noise dosimeters the microphone shall be mounted on the top of the shoulder at the distance of at least 0,1 m from the ear at the side of the most exposed ear and 0,04 m above the shoulder. When using sound level meter, preferably the microphone shall be placed at the centre plane of the worker's head, on a line with the eyes, with its axes parallel to the worker's vision, without the worker present. When worker has to be present, the microphone shall be placed at a distance between 0,1 and 0,4 m from the ear at the side of the most exposed ear. When the worker's position is not well defined, microphone heights that may be used are 1,55 m ± 0,075 m above the ground for standing worker and 0,80 m ± 0,05 m above the middle of the seat plane for seated worker.

By the New Regulation (and the Directive [4]), when applying the exposure limit values, the determination of the worker's effective exposure shall take account of the attenuation provided by the individual hearing protectors worn by the worker ([2], Article 5, paragraph 1 and [4], Article 3, paragraph 2). For the assessment of noise with the hearing protectors worn a method that would comprise of measurements in the ear canal under the hearing protector is highly unrecommended due to the uncertainties caused by the way the protectors are worn by the worker and the technical condition of ear-muffs. The differences in the measured exposures in the same working conditions are up to 30 dB(A) ([15]). The method of choice would be by selection of adequate protectors according to EN 458 [16] depending on measured A-weighted sound pressure levels, sound pressure levels in octave bands and/ or C-weighted sound pressure levels and using HML, octave band or SNR method given by ISO 4869-2 [17] for the estimation of A-weighted sound pressure levels when hearing protectors are worn. However, special measurement procedures are required for the measurement of noise exposure underneath earphones (e.g. for secretaries, telephonists, pilots, air traffic controllers) or underneath helmets (e.g. pilot and motorcycle helmets) ([4], 12.4). For noise sources close to the ear, measurements in the ear canal may be performed in accordance with ISO 11904-1 [18] or ISO 11904-2 [19].

Table 1 *Similarities and differences in instrumentation and measurement methodology by the Old Regulation and the New Regulation (ISO 9612 method).*

Similarities		Differences	
The Old Regulation	**The New Regulation**	**The Old Regulation**	**The New Regulation**
Instrument selection			
sound level meter, integrating sound level meter,	integrating-averaging sound level meter,		
personal dosimeter	personal sound exposure meter		
Instrumentation compliance			
IEC 60804; IEC 60651 Type 1	IEC 61672 Class 1		IEC 61672 Class 2
Calibrator compliance			
IEC 60942 Class 1 (implicitly)	IEC 60942 Class 1		
Periodic verification			
2 years (by other regulation)	2 years		
Noise quantities			
L_{pAeq}	L_{pAeq}	$L_{p,\mathrm{Apeak}}$, $L_{I\ (I=1,..,8)}$	$L_{p,\mathrm{Cpeak}}$
Measurement procedure/strategy and measurement duration			
measurements on all the places of work; shorter measurements providing L_{pAeq} is representative of the cycle of the changes in time	task-based measurement		job-based measurement
longer measurements up to the duration of shift, use of personal dosimeter	full-day measurement		
Field calibration			
before the series of measurements	before the series of measurements		after the series of measurements; discarding of results if the readings before and after differ by 0,5 dB or more
Microphone position			
at the distance of 0,2 m from the ear with worker's presence without the worker's presence 1,6 m above the ground when working standing, 1,2 m above the ground when working seated	at the distance between 0,1 and 0,4 m from the ear at the side of the most exposed ear with worker's presence without the worker's presence 1,55 m ± 0,075 m above the ground when working standing, 0,80 m ± 0,05 m above the seat plane when working seated		when using personal noise dosimeters, on the top of the shoulder at the distance of 0,1 m from the ear at the side of the most exposed ear and 0,04 m above the shoulder measurements in the ear canal underneath the earphones (secretaries, telephonists) or helmets (pilots and motorcyclists)
Other measurement conditions			
full working conditions of work equipment and HVAC	full working conditions of work equipment and HVAC (by other regulation)	measurements when doors and windows closed and if necessary when opened; measurement in all different conditions of machinery operation and tool work if measured levels differ 3 dB or more	special attention to prevent the contributions from non-typical noise sources and non-typical behavior; special attention when treating locations close to noise source and worker moving close to the source

2.2.6. Other measurement conditions

By the Old Regulation, noise in workrooms shall be measured when doors and windows closed, with HVAC turned on, in normal machinery operation and tool work conditions. If the workroom is frequently used with doors or windows opened, the measurements shall be repeated in these conditions. When the noise at the workplace varies during the work time for 3 dB or more, measurements shall be repeated for all the conditions of machinery operation and tool work.

There are no such special requirements in New Regulation, though by more general regulation is requested that the assessment of work environment is made in full working conditions of work equipment and HVAC [20]. By ISO 9612 method, it is essential that the real exposure of the worker is measured. In the cases when possible over- or underestimation is expected (location close to noise source, moving of workers about the machines close to the source) using of personal sound exposure meter is advised. Special attention is to be made to prevent the contributions from non-typical noise sources and non-typical behavior.

3. ASSESSMENT

The assessment is made on the basis of measured noise quantities, by the Old Regulation, equivalent sound pressure levels, L_{pAeq}, sound pressure levels in octave bands, $L_{I\ (I=1,..,\ 8)}$ and A-weighted peak sound pressure level, $L_{p,\mathrm{Apeak}}$, and by the New Regulation equivalent sound pressure levels, L_{pAeq} and C-weighted peak sound pressure level, $L_{p,\mathrm{Cpeak}}$.

3.1. Assessment on the basis of measured equivalent sound pressure levels

Corresponding to the Old Regulation, measured equivalent sound pressure levels are adjusted to rating levels for various types of noise, for impulsiveness, tonality and intermittency.

For impulsive and tonal noise the measured levels are to be adjusted:

$$L_r = L_{pAeq} + K_I + K_T \tag{1}$$

where K_I and K_T are the adjustments for the type of noise, impulsiveness and tonality respectively.

The same adjustments are to be made according to previous version of ISO 9612 ([5], Annex C).

By the Old Regulation (and previous ISO 9612 [5]) the impulse adjustment for the impulsive noise is determined as the difference between the AI-weighted sound pressure level and L_{pAeq}:

$$K_I = L_{pAIeq} - L_{pAeq} \tag{2}$$

By previous ISO 9612, for noise with $K_I \leq 2$ dB the impulse adjustment can be neglected. If K_I is not determined by measurement of AI-weighted sound pressure level, an impulse adjustment of 3dB or 6 dB may be used depending on the prominence of the impulsiveness in the noise (according to previous ISO 9612) and 5 dB (according to the Old Regulation).

As for the current applicable standards, by ISO 7779 ([21], Annex E) the noise is considered to be impulsive if $K_I \geq 3$ dB, calculated according to (2). By DIN 45645-2 [22] the impulse adjustment is:

K_I = 0 dB, if ($L_{p\mathrm{AIeq}} - L_{p\mathrm{Aeq}}$) is less than 3 dB;
$K_\mathrm{I} = L_{p\mathrm{AIeq}} - L_{p\mathrm{Aeq}}$, if 3 dB ≤ ($L_{p\mathrm{AIeq}} - L_{p\mathrm{Aeq}}$) < 6 dB;
K_I = 6 dB, if ($L_{p\mathrm{AIeq}} - L_{p\mathrm{Aeq}}$) ≥ 6 dB.

The tone adjustment for the tonal noise by the Old Regulation is 5 dB. By previous ISO 9612, if tonal components are clearly audible and their presence can be detected by a one-third-octave analysis (that is, if the level of a one-third-octave band exceeds the level of the adjacent bands by 5 dB or more) the adjustment may be 5 dB to 6 dB. If the tonal component is just detectable by the observer a 2 dB to 3 dB adjustment may be applied. As for the current applicable standards, ISO 7779 ([21], Annex D) describes two procedures for determination of whether noise emissions contain prominent discrete tones: the tone-to-noise ratio method and the prominence ratio method, both based on fast Fourier transform (FFT) analysis. ISO 1996-2 [23] gives two objective methods for assessing the audibility of tones in noise, reference method ([16], Annex D) and simplified method ([16], Annex E). Reference method is based on the psychoacoustic concept of critical bands, which are bands defined so that sound outside a critical band does not contribute significantly to the audibility of tones inside that critical band. Method is based on narrow-band frequency analysis (FFT-analysis), it includes procedures for steady and varying tones, narrow-band noise, low-frequency tones, and the result is a graduated adjustment of 0 dB to 6 dB. In the simplified method, for a prominent, discrete tone to be identified as present, the time-average sound pressure level in the one-third-octave band of interest is required to exceed the time-average sound pressure levels of both adjacent one-third-octave bands by some constant level difference. The constant level differences vary with frequency and choices for the level differences are:

- 15 dB in the low-frequency one-third-octave bands (25 Hz to 125 Hz),
- 8 dB in middle-frequency bands (160 Hz to 400 Hz),
- 5 dB in high-frequency bands (500 Hz to 10 000 Hz).

By DIN 45645-2 [22], K_T in (1) is surcharge of tonality and information incorporation. Here surcharge of information incorporation is when person's attention is attracted and person is drawn to listen an unwanted information. The tonality and information adjustment is:

- K_T = 0 dB, if a tone or the noise with information incorporation in noise in working environment is not significant;
- K_T = 3 dB, if a tone or the noise with information incorporation in noise in working environment emerges, but isn't dominant;
- K_T = 6 dB, if a tone or the noise with information incorporation in noise in working environment emerges, and is dominant.

By the Old Regulation, for the intermittent noise two sound pressure levels are determined, first noise level - the equivalent sound pressure level, $L_{pAeq}^{(1)} = L_{pAeq}$, and the second noise level – the equivalent sound pressure level for the periods of highest noise levels, $L_{pAeq}^{(2)}$. For the intermittency measured level is to be adjusted:

$$\boldsymbol{L_r} = \begin{cases} \boldsymbol{L_{pAeq}}^{(2)} - \Delta \boldsymbol{L}\ \boldsymbol{,or} \\ \boldsymbol{L_{pAeq}}^{(1)} \end{cases} \tag{3}$$

whichever amount is the greater. Values for ΔL are given in Table 2.

Table 2 *Adjustments for the intermittency by Old Noise regulation*

Time duration of the interval of highest noise comparing to time duration of noise [%]	50 – 100	25 – 50	10 – 25	10
ΔL [dB]	0	3	6	10

Corresponding to The New Regulation (and the Directive 2003/10/EC whose implementation as well) impulsiveness of the noise is to be taken into account ([2] Article 10, paragraph 1; [3] Article 4, paragraph 6a). However, in the assessment of noise by present version of ISO 9612 [4], unlike the previous version ([5], Annex C) such adjustments are not mentioned. By ISO 9612 from equivalent sound pressure levels noise exposure is calculated:

$$L_{EX,8h} = L_{pAeqT_e} + 10\log\left[\frac{T_e}{T_0}\right] dB \qquad (4)$$

where

$L_{EX,8h}$ is daily noise exposure level;

$L_{p\mathrm{Aeq}Te}$ is the A-weighted equivalent continuous sound pressure level for the effective duration T_e of the working day, and T_0 is the reference duration, T_0=8 h.

$L_{p\mathrm{Aeq}Te}$ is calculated from measured equivalent sound pressure levels depending on measurement strategy chosen (see 2.2.2. and 2.2.3.) and related to this is the measurement uncertainties calculation, to be determined in accordance with ISO 9612 Annex C.

3.2. Presentation of the final results

By the Old Regulation, measured equivalent sound pressure levels (or adjusted rating levels, see 3.1.) shall be rounded to the closest whole number and reported. When on the first decimal place lesser than 5 is that amount is discarded and when 5 or greater is it is rounded to next integer. As for other noise quantities, it is not said but implicitly A-weighted peak sound pressure level should be rounded likewise and sound pressure levels in octave bands should be rounded to one decimal place while they have to be compared to the values rounded in such a way ([1], Table 4).

By DIN 45645-2, measured equivalent sound pressure levels (adjusted rating levels) shall be rounded to the closest whole number and reported. For measurement uncertainty 0 dB or 3 dB is taken, depending on instrumentation class and measurement uncertainty related to task performance (see 2.2.3.). If the instrumentation class is 1 and task performance uncertainty is less than 3 dB, measurement uncertainty shall be 0 dB. If the instrumentation class is 2 and/or task performance uncertainty is less than 6 dB, measurement uncertainty shall be 3 dB. Uncertainty is reported with the final result (e.g.: L_r = (53±3) dB).

By ISO 9612, noise exposure level and C-weighted peak sound pressure level shall be rounded to one decimal place and reported. Uncertainty associated with $L_{EX,8h}$ ([4], Annex C) (and $L_{p,\mathrm{Cpeak}}$, if available) rounded to one decimal place shall be reported as separate value.

4. EVALUATION

Evaluation is made by comparing the final results to the limit values.

The Old Regulation prescribes 5 categories of limit values: regarding risk of hearing damage, regarding interference with work activities, regarding interference with speech communication, regarding indirect communication jamming and regarding interference with sound signal recognition. The New Regulation prescribes limit values regarding risk of hearing damage.

4.1. Evaluation regarding the risk of hearing damage

The Old Regulation prescribes the limit values in this category for measured equivalent sound pressure levels (i.e. rating levels), measured sound pressure levels in octave bands and A-weighted peak sound pressure level.

For equivalent sound pressure levels (i.e. rating levels) limit values are prescribed depending on the daily exposure. Values are given in Table 3 together with the noise exposure levels by ISO 9612 (calculated according to equation (4)) for corresponding time durations and noise levels. There is also the limit of 115 dB that shall not be exceeded for no matter how short the exposition.

Table 3 *Limit values by the Old Regulation regarding the risk of hearing damage and corresponding noise exposure levels by ISO 9612*

daily exposure [h]	The Old Regulation limit value [dB]	daily noise exposure according to ISO 9612 [dB]
8	85	85,0
6	87	85,8
4	90	87,0
3	92	87,7
2	95	89,0
2,5	97	89,7
1	100	91,0
0,5	105	93,0
0,25	110	94,9
0,125	115	96,9

By the Old Regulation, if the value (given in Table (3)) for the measured equivalent sound pressure level (rating level) is exceeded, octave band analysis for the noise is to be done. Limit values for sound pressure levels in octave bands are also prescribed and they are given in form of N-curves ([1], Tables 4 and 6).

As for the $L_{p,\mathrm{Apeak}}$, the limits for daily number of impulses the worker is exposed are prescribed, 100 for sound pressure level of 140 dB and 1000 for sound pressure level of 130 dB.

The New Regulation prescribes two limit values for ISO 9612 noise quantities, $L_{EX,\,8h}$ and $L_{p,\mathrm{Cpeak}}$. They are called limit values and action values, and they are given in Table 4.

Table 4 *Limit values by The New Regulation*

ISO 9612 noise quantities	limit value	action value
$L_{EX,\,8h}$ [dB(A)]	85	80
$L_{p,\mathrm{Cpeak}}$ [dB(C)]	137	135

Comparing the old limit values with the new ones one can see from Table 3 that exposure levels corresponding to old limit values are 85 to 97 dB(A), depending on exposition duration, and that is well beyond the new limits of 80 dB(A) and 85 dB(A) (see Table 5). As for the limit of 115 dB(A), the limit exposition of 80 dB(A) will be reached in 9 seconds and

85 dB(A) in 30 seconds (according to equation (4)), comparing to 15 minutes of allowed exposition by Old Regulation, and by the Old Regulation forbidden level of 115,5 dB(A) the new limits will be reached in 8 seconds (the action level) and 25 seconds (the limit level), so theoretically old limits may be exceeded at least for the very short expositions.

By ISO 9612 methodology it is implied that while assessing A-weighted sound pressure levels the frequency dependence of damaging effect of noise is already taken into account. However, the contributions of higher low-frequency noise loads are increasingly underestimated systematically by the A-weighting, and considering the evidence of its harmful effects (see 4.2.3), and the fact that secondary and tertiary sound protection measures (insulation and damping as well as personal hearing protection) are much less effective at low-frequency sound than at high-frequency noises, low-frequency sound should require particular attention.

As for the limits for peak sound pressure values, it is difficult to compare the limits for A-weighted values to limits for C-weighted values. Depending on the frequency content, levels with low-frequency loads will lead to higher C-weighted comparing to A-weighted sound pressure levels, and ones with high-frequency loads to higher A-weighted comparing to C-weighted levels. So, in practice, depending on noise frequency spectrum the difference of a smaller or greater number of decibels for the values of A-weighted and C-weighted peak sound pressure levels is to be expected. On the other hand, the new limits do not consider possible cumulative effect of a multitude of pulses.

4.2. Evaluation regarding interference with work activities

4.2.1. The Old Regulation's equivalent sound pressure levels (rating levels) limits

The Old Regulation prescribes for equivalent sound pressure levels (i.e. rating levels) three categories of limit values regarding the interference with work activities, depending the type of the source of noise:

a) the working tools or work equipment worker is working with,
b) the working tools the or work equipment worker is not working with (i.e. from a neighboring workplace), and
c) not from the proper production capacities originated (HVAC, traffic, neighboring enterprise). In the Table 5 these limits are listed.

4.2.2. ISO 11690-1 and VDI 2058-3 rating levels limits

While considering noise values that are assessed exclusively for the evaluation regarding the risk of hearing damage, $L_{EX,8h}$ and $L_{p,\mathrm{Cpeak}}$, The New Regulation doesn't give any means for evaluation regarding interference with work activities. As for the current applicable standards, in Table 6 are given the guide limit values for rating levels (calculated according to (1), in terms of DIN 45645-2 [22]), adopted from ISO 11690-1 [24] and VDI 2058-3[25].

4.2.2.1. Predominantly mental activities, activities that demand rating levels less than or equal to 55 dB(A)

Noise influences (more or less impairs) general perceptiveness, attentiveness and concentration, memory, learning ability, reaction capacity, ability to respond, creativity, speech communication. Mental activities are particularly impaired if noises exhibit mayor fluctuations or have an informative content, e.g. linguistic information. Steady noise impairs by masking verbal intelligibility at low speech levels or at a distance.

Table 5 *Limit values by The Old Regulation regarding the interference with work activities depending the type of noise source*

Type of work activity	Limit values by the type of noise source [dB]		
	a	b	c
Physical work with no need for mental concentration and observations by hearing	85	85	80
Physical work concentrated on accuracy; periodical control of working environment by hearing; driving and controlling the transport vehicles and transportation means	80	75	70
Work under frequent voice command and acoustical signals Work demanding permanent observations of environment by hearing Routinely work activities of predominantly mental type	75	70	60
Routinely work activities of predominantly mental type that demand concentration	70	65	55
Mental work focused on work control of group of people that performs mainly physical work Work demanding concentration or direct communication by means of speech and telephone	-	60	50
Mental work focused on work control of group of people that performs mainly mental work Work demanding concentration and direct communication by means of speech and telephone Work exclusively by means of communication (telephone, etc.)	-	55	45
Mental work demanding great concentration, the exclusion from surrounding, precise psychomotor work or communication with a group of people	-	-	40
Mental work as creating concepts, work related to great responsibility, communication to reaching an agreement with a group of people	-	-	35
Concert halls and theatres	-	-	30

Table 6 *Guide values for limits for work activity categories rating levels (adopted from ISO 11690-1 and VDI 2058-3)*

Work activity category	Guide limit value for rating level [dB]
Predominantly mental activities	≤ 55
Simple or practiced office activities and comparable activities	≤ 70
other activities	> 70

Activities that are considerably influenced (more or less impaired) by noise are the following.

Activities requiring creative thought under major time pressure, e.g. editing or authoring scientific, journalistic or artistic works. Noise loads during such activities shall be avoided as greatly as possible, including those with rating level significantly below 55 dB(A).

Activities with attention to a work object in conjunction with important decisions. If the activities are based on spoken language, verbal intelligibility and shielding against distractive noise including irrelevant speech are required.

Activities characterized by inner talking, e.g. understanding complex text with complicated sentence structures, translating foreign languages, monitoring workflows. Irrelevant background speech, highly segmented background music and irregular or impulsive background noises (rattling, ringing) shall be avoided.

Such activities typically occur:

- during attendance at discussion and negotiations in conference rooms;
- when reading in rooms of libraries;
- during examinations in educational establishments;
- during work and decision-making processes (possibly under time pressure) in offices;
- during medication examinations, treatments and operations;
- in monitoring and control rooms during the monitoring and control of process sequences;
- during drafting, translation, dictation, noting down and correction of difficult texts.

4.2.2.2. Office and comparable activities, activities that demand rating levels 55 dB(A) to 70 dB(A)

This is for the activities that are not associated with a continuously high strain on attentiveness (concentration), that is when the activity includes routine elements, recurrent similar and easy to process tasks or work contents. Activities that are not characterized by high decision-making pressure. Activities that require good verbal intelligibility require rating level significantly below 70 dB(A).

Activities in those rating level of 70 dB(A) should not be exceeded are:

- activities characterized by informative and communicative content, such as serving and advising customers in department stores or similar establishments;
- work characterized by psychomotor (fine-motor) activity (eye-hand coordination);
- planning, data acquisition in text processing/ data processing;
- work in offices and laboratories;
- supervision, control and monitoring activities in instrument and process control rooms;
- selling, attending to customers, activities with official opening hours;
- difficult precision assembly work.

4.2.2.3. Activities with rating levels greater than 70 dB(A)

Activities at workplaces where an A-weighted rating level of maximum 70 dB frequently cannot be complied with when taking reasonable operational noise-inhibiting measures into consideration, are e.g.:

- skilled manual activities (production, installation);
- activities on production machines, equipment, devices;
- maintenance, repair and cleaning of technical equipment and direct supervision thereof;
- operation of processing machines for metal, wood and similar.

4.2.3. Octave band analysis and frequency and time dependent noise loads

By the Old Regulation, same as in evaluation regarding the risk of hearing damage, if the value (given in Table 5) for the measured equivalent sound pressure level (rating level) is exceeded, octave band analysis for the noise is to be done. Values in octave bands shall be compared to the limit values that are given ([1], Table 4).

As already said (see 4.1.) in ISO 9612 methodology it is assumed that while assessing A-weighted sound pressure levels the frequency dependence of damaging effect of noise is already taken into account.

However, the noise loads of the same A-weighted sound pressure levels with different frequency accentuated energy components (e.g. high-frequency or low-frequency accentuated or narrow-band sound immissions), as well as time fluctuating sound are experienced very differently, from dull, more pleasantly evaluated noise (low-frequency accentuated) to sharp (high-frequency sound). Low-frequency noise loads (octave band noise around 250 Hz) lead to lower strains on the hearing than high-frequency loads (octave band noise around 2000 Hz). A-filter attenuation reduces even significant low-frequency components of noise to negligible amount, and high-frequency noise components are also underrated.

Stationary low-frequency noises such as uniform humming in conjunction with a monotone activity (like a monotone monitoring activity) can cause drowsiness and absent-mindedness. Low-frequency noises cause a specifically straining effect at low levels, in particular if the energy or power is highly concentrated on the frequency range below about 100 Hz. Amongst other symptoms, lack of concentration, fatigue, headache, feelings of anxiety and nervousness occur, and this is associated with a significant restriction in performance and extended reaction times. In DIN 45680 [26] a method is given for measuring and assessing of low-frequency noise. Evaluation is made for noise components in the frequency range from 8 Hz to 100 Hz in case when the difference between the C-weighted and A-weighted level exceeds 20 dB (indication for significant energy concentration below 100 Hz).

A method similar to comparing octave bands levels to limit values by the Old Regulation is ANSI S12.2 [27] NC method. In this standard the noise level criteria on the basis of the space use are given ([26], Annex C, table C.1) and two methods of rating that involve octave band analysis, method that employs noise criteria (NC) curves ([26], Table 1 and Figure 1) and the method for evaluating low-frequency fluctuating noise using room noise criteria (RNC) curves ([26], Table 2 and Figure 4). Recommended NC and RNC criteria are given ([26], Annex C, table C.2).

The NC procedure is the following. From measured sound pressure levels in octave bands 250 Hz, 500 Hz, 1000 Hz and 2000 Hz SIL index is calculated (L_{SIL} in (5), see 4.3.). Octave levels spectrum is plotted on the NC curves diagram. If the spectrum doesn't exceed the NC curve for the calculated SIL (NC-SIL value), the rating level is SIL value. If it exceeds it in any octave band, the tangency method is used to find NC level (rating level). The rating level is the highest NC curve value that the plotted spectrum touches. If the plotted spectrum differs for more than 9 dB from NC curve with the calculated SIL value (NC-SIL curve) in octave bands below 500 Hz that would lead to serious dissatisfaction due to low-frequency noise. The method is presented in Figure 3.

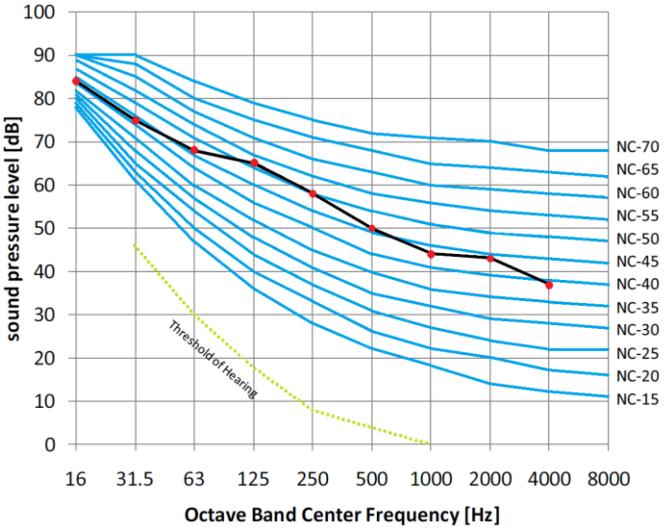

Fig. 3 *The NC method for the octave level spectrum with SIL= 49 dB. The rating level is NC-51. Spectrum exceeds NC curve at low frequencies but only by 2 dB so there is a little chance of serious dissatisfaction*

4.3. Evaluation regarding interference with speech communication and indirect communication jamming

In Table 7. the limit values by The Old Regulation regarding the interference with speech communication and indirect communication jamming ([1], Tables 2 and 3) are given.

Table 7 *Limit values regarding the interference with speech communication and indirect communication jamming by the Old Regulation*

Measured (rating) level [dB(A)]	Possibility of indirect communication (e.g. by phone)	Distance [m] allowing direct speech communication	
		normal voice	loud voice
45	satisfactory	7	14
50	satisfactory	4	8
55	satisfactory	2,2	4,5
60	little bit difficult	1,3	2,5
65	little bit difficult	0,7	1,4
70	hard	0,4	0,8
75	not satisfactory	0,22	0,45
80	not satisfactory	0,13	0,25
85	not satisfactory	0,07	0,14
90	not satisfactory	-	0,08

As for the current applicable standards, the criteria for speech intelligibility are given in ISO 9921 [28]. From measured sound pressure levels in octave bands 500 Hz, 1000 Hz, 2000 Hz and 4000 Hz the speech interference level of noise is calculated (L_{SIL})

$$L_{SIL} = \frac{1}{4}\sum_{i=1}^{4} L_{N,oct,i} \tag{5}$$

For the indirect communication (in ISO 9921 defined as personal communication system: intercom, telephone and mobile system) speech transmission index, STI is calculated by method given ([28], Annex C). The intelligibility ratings are given in Table 9.

The speech interference level (SIL) is given by difference between speech level ($L_{S,A,L}$), that is A-weighted sound pressure level of speech at a distance L (in meters) from speaker, and speech interference level of noise. Speech interference level is related to vocal effort of speaker and the relation is given in Table 8 for the distance of 1m.

Table 8 *Vocal effort of a speaker and related A-weighted speech level at 1m in front of the mouth (ISO 9921)*

Vocal effort	$L_{S,A,1m}$ [dB]
Very loud	78
Loud	72
Raised	66
Normal	60
Relaxed	54

For the speech levels above the level of $L_{S,A,1m} > 75$ dB there is also decrease of speech quality with loud speech to be included. Value of $L_{S,A,1m}$ in Table 8 shall be reduced by 0,4($L_{S,A,1m}$-75). For the situation of wearing hearing protector for the noise levels above the level of $L_{S,IL} > 75$ dB the values of $L_{S,A,1m}$ in Table 8 shall be reduced by 3 dB.

The speech interference level at a distance L may be determined from equation:

$$L_{S,A,L} = L_{S,A,1m} - 20\log L \tag{6}$$

Fair speech-communication intelligibility is ensured if SIL= $L_{S,A,L}$- $L_{SIL} \geq 10$ dB. Intelligibility ratings for SIL index are given in Table 9.

A reduced intelligibility is observed with non-native but fluent speakers and listeners of a second language. For such situation to SIL value 4 dB should be added.

Table 9 *Intelligibility rating and relation with SIL and STI indices (ISO 9921)*

Intelligibility rating	SIL [dB]	STI
Excellent	21	> 0,75
Good	15 to 21	0,60 to 0,75
Fair	10 to 15	0,45 to 0,60
Poor	3 to 10	0,30 to 0,45
Bad	< 3	< 0,30

Using this method, three examples may be listed for different requirements and talk situations.

For highly demanding activities (perfect intelligibility of speech, relaxed/normal voice) with a distance of 2 m to 4 m between the interlocutors, the A-weighted sound pressure level should not exceed 30 dB to 40 dB.

For less demanding activities (good intelligibility of speech; normal/raised voice) with a distance of 1 m to 2 m between the interlocutors, the A-weighted sound pressure level of the extraneous noise should not exceed 45 dB to 55 dB.

For lower requirements (satisfactory intelligibility of speech; raised voice) with a distance of 1 m to 2 m between the interlocutors, the A-weighted sound pres-sure level of the extraneous noise should not exceed 55 dB to 65 dB. This requirement is only practical for short talks.

For the indirect communication (in ISO 9921 defined as personal communication system: intercom, telephone and mobile system) speech transmission index, STI is calculated by method given ([28], Annex C). The intelligibility ratings are given in Table 9.

4.4. EVALUATION REGARDING INTERFERENCE WITH SOUND SIGNAL RECOGNITION

By The Old Regulation noise level at the place of worker shall be minimum 10 dB(A) lower than level of the alarm signal ([1], article 15.). By ISO 7731 [29] three measurement methods for the assessment of the auditory danger signals are given. To ensure its audibility, the A-weighted sound-pressure level of the danger signal shall not be lower than 65 dB at any position in the signal reception area, and in addition, at least one of the following criteria shall be met.

For measurements of the A-weighted sound-pressure level (method a)), the difference between the two A-weighted sound-pressure levels of the signal and the ambient noise shall be greater than 15 dB.

For measurements of the octave-band sound-pressure level (method b)), the sound pressure level of the signal in one or more octave-bands shall exceed the effective masked threshold by at least 10 dB in the octave-band under consideration.

For measurements of the 1/3 octave-band sound-pressure level (method c)) the sound pressure level of the signal in one or more 1/3 octave-bands shall exceed the effective masked threshold by 13 dB in the 1/3 octave-band under consideration.

The procedure for calculation of effective masked threshold is given ([29], Annex B). When measuring octave-band sound pressures (method b)) for the band under consideration is the maximal value of level in octave band and level in neighboring octave-band reduced by 7,5 dB. When measuring 1/3 octave-band sound pressures (method c)) for the band under consideration is the maximal value of level in 1/3 octave band and level in neighboring 1/3 octave-band reduced by 2,5 dB.

5. CONCLUSION

With the entry into force of the New Regulation the method of noise assessment at workplace and the measurement instrumentation requirements comparing to the Old Regulation rest mainly the same, though the proscribed ISO 9612 measurement method is designed to assess noise exposure levels for the evaluation regarding the risk of hearing damage. As for evaluation by comparing the final results to the limit values, The New Regulation prescribes only limit values regarding the risk of hearing damage. Comparing these values to Old Regulation limit values one can see that in most cases when new limits are not exceeded, neither are the old ones (see 4.1.), and that is in compliance with Directive 2003/10/EC whose national implementation New Noise Regulative is. However, there rests a great gap in the assessment and evaluation of workplace noise for the level range underneath the risk of hearing damage, the domain that the Old Regulation covered well. As the assessment method in addition to ISO 9612 method, DIN 45645-2 method for the determination of the noise rating level for occupational activities at the work place is proposed, and for the limit values for the evaluation regarding interference with work activities VDI 2058-3 and ISO 11690-1 rating levels limits. In addition, the methods according to existing standards are given for evaluation regarding interference with speech communication, indirect communication jamming and interference with sound signal recognition, also covered before by the Old Regulation.

REFERENCES

[1] Regulation on occupational safety measures and norms for noise protection at workplace. Off. J. SFRJ. 1992, 21, pp. 310-316

[2] Regulation on preventive measures for health and safety at work regarding noise exposure. Off. J. RS. 2011, 96, pp. 10-12 and Off. J. RS. 2015, 78, p. 24

[3] Directive 2003/10/EC of the European Parliament and of the Council of 6 February 2003 on the minimum health and safety requirements regarding the exposure of workers to the risks arising from physical agents (noise). Off. J. Eur. Commun. L 42, 15.2.2003, p. 38

[4] ISO 9612:2009, SRPS EN ISO 9612:2016 Acoustics - Determination of occupational noise exposure - Engineering method

[5] ISO 9612:1997, SRPS EN ISO 9612:2008 Acoustics - Guidelines for the measurement and assessment of exposure to noise in a working environment

[6] IEC 60651:1979+AMD1:1993+AMD2:2000 Sound level meters

[7] IEC 60804:2000 Integrating-averaging sound level meters

[8] IEC 61672-1:2002 Electroacoustics - Sound level meters - Part 1: Specifications

[9] IEC 61672-2:2003 Electroacoustics - Sound level meters - Part 2: Pattern evaluation tests

[10] IEC 61252:1993+AMD1:2000 Electroacoustics - Specifications for personal sound exposure meters

[11] IEC 60942:2003 Electroacoustics - Sound calibrators

[12] JUS N.R6.032:1982 Precision sound level meters

[13] IEC 60179:1973 Precision sound level meters

[14] Regulation on the types of measurement instrumentation for those that the verification is mandatory and on the time intervals of their periodic verification. Off. J. RS. 2010, 49, p. 45 and Off. J. RS. 2013, 110, pp. 210-211

[15] Kotarbińska E. and Kozłowski E. Measurement of Effective Noise Exposure of Workers Wearing Ear-Muffs. International Journal of Occupational Safety and Ergonomics (JOSE). 2009, 15(2):193–200.

[16] EN 458:2004, SRPS EN 458:2008 Hearing protectors - Recommendations for selection, use, care and maintenance - Guidance document

[17] ISO 4869-2:1994+Cor 1:2006, SRPS EN ISO 4869-2:2012 Acoustics - Hearing protectors - Part 2: Estimation of effective A-weighted sound pressure levels when hearing protectors are worn

[18] ISO 11904-1:2002, SRPS EN ISO 11904-1:2012 Acoustics - Determination of sound immission from sound sources placed close to the ear - Part 1: Technique using a microphone in a real ear (MIRE technique)

[19] ISO 11904-2:2004, SRPS EN ISO 11904-1:2012 Acoustics - Determination of sound immission from sound sources placed close to the ear - Part 2: Technique using a manikin

[20] Regulation on the procedure of inspection and checking of work equipment and examination of work environment. Off. J. RS. 2006, 94, pp. 10-13, 2006, 108, p. 15 , 2014, 114, pp. 24-25 and Off. J. RS. 2015, 102, p. 34

[21] ISO 7779:2010, SRPS EN ISO 7779:2012 Acoustics - Measurement of airborne noise emitted by information technology and telecommunications equipment

[22] DIN 45645-2: 2012 Determination of rating levels from measurement data - Part 2: Determination of the noise rating level for occupational activities at the work place for the level range underneath the given risk of hearing damage

[23] ISO 1996-2:2007, SRPS ISO 1996-2:2010 Acoustics - Description, measurement and assessment of environmental noise - Part 2: Determination of environmental noise levels

[24] ISO 11690-1:1996, SRPS EN ISO 11690-1:2012 Acoustics - Recommended practice for the design of low-noise workplaces containing machinery - Part 1: Noise control strategies

[25] VDI 2058-3:2014 Assessment of noise in the working area with regard to specific operations

[26] DIN 45680:1997 Measurement and assessment of low-frequency noise immissions in the neighbourhood

[27] ANSI/ASA S12.2-2008 Criteria for Evaluating Room Noise

[28] ISO 9921:2003, SRPS EN ISO 9921:2008 Ergonomics - Assessment of speech communication

[29] ISO 7731:2003, SRPS EN ISO 7731:2010 Ergonomics - Danger signals for public and work areas - Auditory danger signals

CONTROL OF NOISE LEVEL IN THE KINDERGARTEN "LJUBICA POPOVIC" PODGORICA

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Abstract – *The children are vulnerable population group that are subject of a special regime of ensuring the appropriate conditions for them in order to protect their lives and health in premises in which they reside. Because of their undeveloped immune systems, children who reside in kindergarten, are especially vulnerable. That is especially because children spend at least 4 to 8 hours in kindergartens. In addition to the quality of nutrition, education and medical staff, good hygienic conditions, microclimate conditions and noise levels in residential areas, health sanitary recommendations of 10 m² for babies, and 6-8 m² for other children in kindergarten according to the norms of the number of children in the group prescribed by the Montenegrin law on pre-school Education. The kindergarten educational units (9 of them) of "Ljubica Popovic" that takes place in Podgorica accept about 3,000 children aged 1-7 years involved and various forms of activates. This year the Institute of Public Health conducted the measurement of noise levels in classrooms and playgrounds of these educational units. The results show that the noise level has is in the limits of statutory value in classrooms, also in the measured noise level on the playgrounds of kindergartens.*

Keywords: *kindergarten, children, noise level.*

1. INTRODUCTION

Noise is a subjective unpleasant listening experience, ubiquitous adverse factor in the environment and is among the physical factors detrimental to health. In European Union the noise in the environment is one of the leading environmental problems. According to the World Health Organization, about 120 million people have a hearing problem, more than half of Europeans live in a noisy environment, and night noise disrupts sleep one third of Europeans. [6] Noise in urban areas from working and living environment. Noise in the vicinity of schools and kindergartens, as well as the place of residence, is one of the factors that influence blood pressure in children together with factors such as gender, age, nutritional status. Noise interferes with voice communication or voice teachers covering acoustic energy of another audio source. What is a masking sound stronger, the greater the number of words students to be incomprehensible. Traffic noise around schools reduces the student 'ability to read, short and long term memory, often causes headaches and migraine in schoolchildren[3].

The main objective of this paper is to determine and monitor the noise levels in class in kindergartens, protect children health and improvement of residence conditions.

Kindergarten "Ljubica Popović" is located in Podgorica, accept about 3000 children, in nine units, aged second 1 -7 years involved in various forms of activates.

In the framework of the public health control kindergartens, measured the noise in nine residential units of public preschools and theirs playgrounds.

2.METODOLOGY

During the control of sanitary and hygienic conditions in preschools, was done and control of noise levels in classrooms and a playgrounds of kindergardens. Measurement was done in nine units: Sunce, Suncokrili, Ljubica Popović, Lane, Osmjeh, Pčelica,Zvjezdica, Palčica i Bajka and kindergarten playgrounds. Six units are located in residential areas, three in the mixed. It was measured precision modular analyzer Brϋel & Kjaer Model 2250 which meets the prescribed standard of IEC60804. Measurements were done in accordance with standard [4] and the requirements of legislation[7].

Before the measurements was performed calibration, used the wind protector for the outside during the measurements. The minimum duration of measurement interval is 15 minutes.

In the classrooms were measured, temperature, humidity, and outside temperature, humidity, air pressure and velocity. Meteorological conditions were a favorable measurement.

Determination of the equivalent noise level in the environment-outside of kindergardes, was done according to the prescribed methodology, and results were compared with the Regulations on limit values for environmental noise, the method of determining the noise indicators and acoustic zones and assessment methods adverse effects [1].

Table 1. *Limit values of noise indicators indoors [2]*

<table>
<tr><th rowspan="2"></th><th rowspan="2">Purpose room</th><th colspan="2">Noise level in dB (A)</th></tr>
<tr><th>Day</th><th>Night</th></tr>
<tr><td>1.</td><td>Residence rooms (bedroom and living room) in a building with windows closed</td><td>35</td><td>30</td></tr>
<tr><td>2.
2.1</td><td>The public and other buildings, with windows closed:
Health institutions and private practice, and in them:</td><td>35</td><td>30</td></tr>
<tr><td></td><td>a) Hospital room</td><td>40</td><td>40</td></tr>
<tr><td></td><td>b) Clinics</td><td>35</td><td>35</td></tr>
<tr><td></td><td>c) The operational block without medical devices and equipment</td><td>35</td><td>30</td></tr>
<tr><td>2.2</td><td>Rooms in facilities for children and students, and a bedroom homes, facilities in facilities for children and students home, and bedroom homes for a stay of elderly and pensioners</td><td>35</td><td>30</td></tr>
<tr><td>2.3</td><td>Rooms for educational work (classrooms, lecture hall, classrooms, etc.), Movie theaters and reading rooms in libraries</td><td>40</td><td>40</td></tr>
<tr><td>2.4</td><td>Theater and concert hall</td><td>30</td><td>30</td></tr>
<tr><td>2.5</td><td>Hotel rooms</td><td>35</td><td>30</td></tr>
</table>

The measured noise levels in kindergarten units is done in empty classrooms, in accordance with the recommendations of standard [4]. We could not compare with the current legislation in Montenegro, because it does not currently exist, so the results for the purposes of scientific research, commented in relation to the current legislation of Serbia [2], where the prescribed limit value of identical noise levels prescribed by law in other countries of the region. Table 1. date limit values of noise indicators indoors under Regulation [2].

3.RESULTS

Analising obtained value of equivalent noise level, which were measured in kindergardens "Ljubica Popović", we are trying to demonstate the real case: does equivalent noise level exceed the law limitation for noise in environment, if it exceeds, we will try to answer the question what causes that. All measurements were done in day interval of measurements, after the regular activities of kindergardens because the purpose was to find out the real state on site.

In Table 2. are showen the results of measures of equivalent noise level L_{eq} in duration of 15 minutes in accordance with the guidelines 1996-1, 1996-2, in kindergardens "Ljubica Popović" during the september 2016, in nine educational units, ten classrooms and nine playgrounds of "Ljubica Popović".

As shown in Table 2 the equivalent noise level measured in all classrooms no exceed the regulations limits for rooms in facilities for children in accordance with limit by Guidelines [2]. The measured values were from 29.5 to 32.0 below the provided by law limit of 35 dB. Exception is classroom 1 in unit “Pčelica”, where is measured noise level 41.5 dB, this classroom is on the opposite side where is main city street.

Law predicted limits for noise levels from environment, were not exceed on all playgrounds. These values are the result of the position of the kindergarten located in a residential zone, except units „Ljubica Popović“ „Pčelica“ and „Osmjeh“ located in the mixed zone. Facilities in the residential zone are far from the local road, with large yards where there is only the influence of the usual activities in residential zones.

Table 2. *The reasults of measures taken in kindergarten “Ljubica Popović” in nine educational units in September 2016.*

Kindergarten “Ljubica Popović”	Class L_{Aeq} [dB]	Allowed*	Outside of kindergarten L_{Aeq} [dB]	Allowed
Unit 1 – “Ljubica Popović”	30.5	35	52.3	55
Unit 2 – “Sunce”	29.7	35	49.5	55
Unit 3 – “Suncokrili”	31.0	35	51.8	55
Unit 4 – “Pčelica” Class 1	41.5	35	51.3	60
Unit 4 – “Pčelica” Class 2	31.5	35	-	60
Unit 5 – “Lane”	31.0	35	48.9	55
Unit 6 – “Zvjezdica”	32.0	35	49.4	55
Unit 7 – “Osmjeh”	31.0	35	54.5	60
Unit 8 – “Bajka”	32.0	35	50.5	55
Unit 9 – “Palčica”	29.5	35	48.7	55

4.CONCLUSION

The results of measurements of noise levels in classrooms kindergartens show that levels do not exceed the legally defined noise levels in the environment, except in classroom 1 in unit “Pčelica”, this classroom are on the opposite side where is main city street. It is recommended for classrooms in the unit "Pčelica" which is measured by an increased level of noise that to change old wooden windows with new anodized.

The results of measurements of noise levels on playgrounds kindergartens show that levels do not exceed the legally defined noise levels in the environment. These results indicates that children who reside in kindergartens are not exposed to increased level of noise from the environment, conditions are in accordance with the recommendations[5] and no negative impact on theirs activities in kindergartens, growth and development. This indicates that children staying in kindergartens are not exposed to an increased level of noise from the environment and that there is no negative impact on their growth and development, dont exist negative impact of high level of noise.

REFERENCES

[1] Low on Environmental Noise Protection *(Off. Gazette of Montenegro no. 28/2011).*

[2] Rulebook on limit values for environmental noise, the method of determining the noise indicators and acoustic zones and methods of assessment of harmful effects *("Off. Gazette of Montenegro", no. 60/11)*

[3] Regulation on noise indicators , limit values, methods for assessing indicators of noise , disturbance and adverse effects of environmental noise *("Off. Gazette of the RS ", no . 75/2010).*

[4] World Health Organization. *"Guidelines for community noise", WHO, Geneva, 1999*.

[5] World Health Organization, Regional Office for Europe and JRC European Commission*., "Burden of disease from environmental noise",. Quantification of healthy life years lost in Europe. WHO, Copenhagen, 2011*.

[6] J.Jorga, Hygiene with medical ecology, *Faculty of Medicine, Belgrade, 2014, p.68-69.*

[7] MEST EN ISO 1996-1, MEST EN 1996-2

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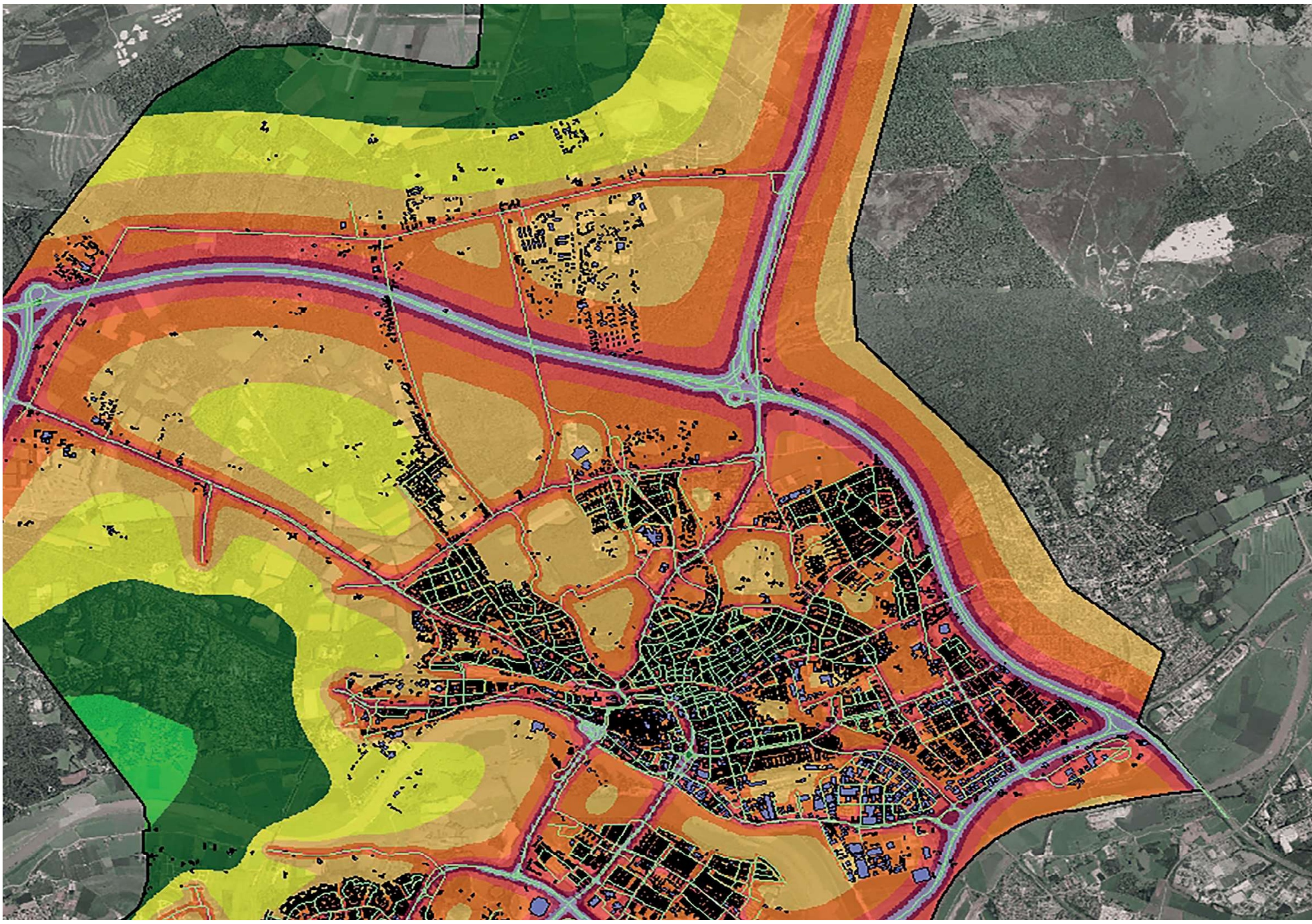

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BEYOND MEASURE

NOISE LEVEL MONITORING IN HOSPITAL ZONE

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Abstract – *Noise in the hospital zones is contributing significantly to overall noise levels and is one of the major factors that impede recovery of patients. World Health Organization document "Night noise guidelines for Europe" indicate that maintaining an equivalent level of noise up to 30 dB LAeq is necessary to allow adequate rest and recovery of patients. The relevant legislation in Montenegro classified the hospital zones as zones of increased regime of protection against noise. Therefore, the statutory noise level in decibels (A) ranges from 50 dB for the day and evening to 40 dB for night interval.*

The aim of this study was to analyze data on the level of noise pollution in the hospital area of the Clinical Center of Montenegro. This center represents a unique medical institution that provides health services to the public at the tertiary level of health care for the entire Montenegro, and at the secondary level as a general hospital for municipalities Podgorica, Danilovgrad and Kolašin. Level of environmental noise plays very important role affecting largely outcomes of treatments.

Institute of Public Health did a seven-day monitoring of noise in the hospital zone of the Clinical Center of Montenegro. Measured noise levels were significantly above the statutory levels and WHO recommendations.

Keywords*: hospital zone, patients health, increased noise levels.*

1. INTRODUCTION

Noise has become ubiquitous ecological factor. Noise have a negative impact on hearing, on the intellectual work, mental health, have contributions for stressful effects, especialy especially on the cardiovascular system, etc. Every fourth resident in European countries exposed to noise above 55 dB [5]. Twenty percent live in black acoustic zones with daily noise levels above 65 dB.Trećina European population is complain about the disruption of sleep due to nighttime levels above 45 dB. Noise pollution causes 43.000 hospital admissions in Europe per year [5]. World Health Organization indicate that maintaining an equivalent level of noise up to 30 dB LAeq is necessary to allow adequate rest and recovery of patients [3]. Among the many negative psychological consequences that the population of endangered municipal noise can be expected is scrambling to sleep are the basic and most important. Intermittent noise has negative effects on sleep from the constant, especially in periods of deep sleep. In the current field and laboratory studies has been shown that noise increases the time required to fall asleep, sleep seems superficial and leads to frequent waking. After waking effects are manifested in the form of fatigue, mood changes [2]. It is particularly sensitive to vulnerable groups of people, patients in hospitals. In this paper we will review the results of an experiment we conducted on the level of noise in a hospital zone.

2. METODOLOGY

Noise level measurement is done in a hospital zone, zone under increased protection regime of noise at the measuring point in front of the Clinical Center of Montenegro in Podgorica. For this experiment was selected as one position, characterized by various sources of noise, and recognize the noise of the cars, motorcycles, patients, health care workers (Figure 1).

The measuring position is located inside the circle Clinical center of Montenegro, with local road which is characterized by heavy traffic especially in the daily interval measurements, which is located at a distance of more than 50 meters of the measuring point, also, the large influx of people into the daily interval measurements present due to medical examinations in Clinical center of Montenegro, as well as the presence of emergency vehicles that transported patients from other health centers of the country for treatment as Clinical center of Montenegro.

The measurements were made in the seven-day period from 30.05 to 05.06.2014. in the reference interval for the day (evening) and night for 15 minutes.

The equipment was used for this measurement is accurate modular analyzer 2250 Bruel & Kjaer producers who meet the required standards IEC60804.

The measurements were carried out in accordance with MEST EN ISO 1996-1 and MEST EN ISO 1996-2 and the requirements of legislation [4]. Before the calibration measurements, and used the wind deflector during interval measurements. Equipment for measuring noise is set at a height of 1.5 m on a hard surface. Working conditions were favorable to the standards recommended [6]. Determination of the equivalent noise level was conducted in accordance with the prescribed methodology, and results were compared with the Rulebook on limit values of environmental noise, the method of determining the noise indicators and acoustic zones and methods of assessing adverse effects [1].

Valid legislation in Montenegro puts the hospital zones in areas which are subject to high security mode noise and which is necessary to protect particularly vulnerable groups of the population, but the new statutory noise in decibels (A) ranges from 50 dB for the day and evening to 40 dB for night interval [1].

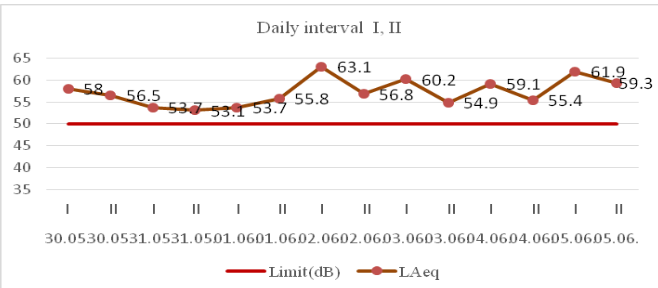
Figure 1. *Measuring point*

3. RESULTS OF EXPERIMENT

During the seven days were carried out measurements of noise levels at selected measuring position in five different measurement terms for a period of 15 minutes. The results of the measurements for all measurement intervals are shown in table 1 and graphs (Graf 1, 2 and 3).

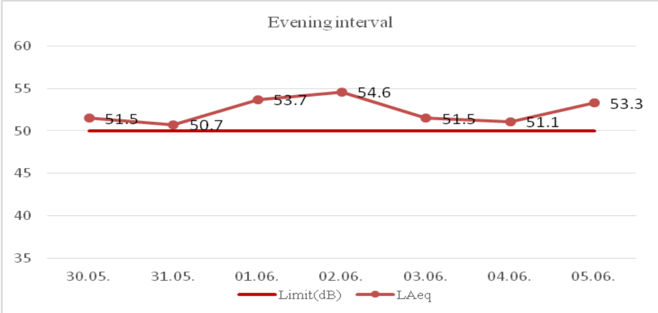
Table 1 *The results of the measurements for all measurement intervals*

INTERVAL	DAILY I	DAILY II	EVENING	NIGHT I	NIGHT II
LIMIT	50	50	50	40	40
30.05.2014	58.0	56.5	51.5	51.5	50.1
31.05.2014	53.7	53.1	50.7	55.7	45.5
01.06.2014	53.7	55.8	53.7	46.3	48.6
02.06.2014	63.1	56.8	54.6	49.9	50.4
03.06.2014	60.2	54.9	51.5	47.7	56.8
04.06.2014	59.1	55.4	51.1	51.7	51.5
05.06.2014	61.9	59.3	53.3	48.1	53.9



Graf 1. *The results of daily measurement intervals I, II of noise levels for the period 30.05- 05.06.2014.g*

Graph 1 show the daily measurement intervals, was record exceeding the levels, in relation to the permitted value equivalent level of noise in the selected position. The deviation from the allowed values are ranging from 1 dB to 13 dB, while the evening interval exceeding the range from above 1 dB to 5 dB (graph 2).

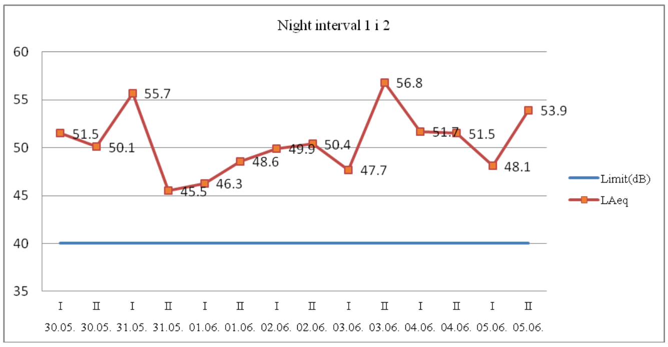


Graph 2. *The results of evening measurement intervals of noise levels for the period 30.05- 05.06.2014.g*

The graph 1 presents the measured values of noise in the daily interval measurements on the measuring position in front of the Clinical Centre of Montenegro in Podgorica. The consequence of that is the presence of a large number of people during hospital examinations in clinics Clinical center of Montenegro, as well as visits to patients staying in the hospital, intensive traffic on the local road is the main reason for such a high measured values of noise levels in this area.

In the evening the interval measurements were also recorded exceeding the limit noise levels, exceeding the range from 1 dB to 5 dB, as a result of the reduced number of people within Clinical center of Montenegro and vehicles on local roads in the measurement interval.

The graph 3 show deviations recorded in the night intervals ranging from 5 dB to 17 dB. The main sources of noise are identified vehicles running local road, the outdoor air conditioner on Clinical center of Montenegro and sporadic murmur of people in front of Clinical centre of Montenegro.



Graph 3. *The measurement results for the night measurement intervals I, II for the period 30.05 to 05.06.2014*

4. CONCLUSION

Experimental results indicate the presence of increased noise environment inside the circle Clinical center of Montenegro, in a hospital zone, a zone that falls within the zone of enhanced protection regime of noise.

The vulnerability of patients is evident during the day and especially at night time when it is essential quiet and continuous dream for faster recovery recumbent patients.

Overdrafts noise levels range from 1 dB to 13 dB in the daily interval measurements, while the night interval exceeding the range from 5 dB to 17 dB. The measured values indicate the presence of problems which definitely represents a disruptive factor which can cause pain, insomnia and to reduce the pace of recovery of the patient. What kind and how much negative impact on the recovery of this vulnerable category of the population, we can only assume.

This problem needs to be addressed in order to find the best solution for reducing noise levels at the statutory level, either at local level or to independent actions in suppressing noise, because in this way solve the problem in those health institutions in Montenegro that have a similar problem.

5. REFERENCE

[1] Rulebook on limit values of environmental noise, the method of determining the noise indicators and acoustic zones and methods of assessment of adverse *effects ("Off. Gazette of Montenegro", no. 60/11)*

[2] J.Jorga, Hygiene with medical ecology, *Faculty of Medicine, Belgrade, 2014*

[3] Night noise guidelines for Europe, *WHO, 2009.*

[4] Rulebook on the methods of calculation and measurement of the noise level in the environment *(Official Gazette of Montenegro 27/14)*

[5] EEA *Report "Noise in Europe 2014", www.eea.europa.eu/noise*

[6] MEST EN ISO 1996-1 and MEST EN ISO 1996-2

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ADJECTIVAL COMPOUNDS IN ENVIRONMENTAL NOISE DISCOURSE

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Abstract – *Compounding is a very productive way of new word formation in English and scientific discourse is specific in its use. The topic of this paper, adjectival compounds as used in environmental noise discourse, presents one aspect of succint expression of complex meanings.*

1. INTRODUCTION

Environmental Noise Barriers, title of Benz Kotzen and Colin English's book, is informative in two important ways. Essentially, it 'reflects the growing concern of the general public about noise pollution' (Kotzen and English 1999: 1), which has become a serious threat to the living environment owing to the rapid development of road and railway traffic. To deal with the issue and, if not eliminate, mitigate the health-threatening problem, a lot of developed countries across Europe, the USA, Australia and the Far East have acknowledged the need for a quieter environment to be achieved by the introduction and clever design of physical shields or barriers. In the other, linguistic respect, the noun phrase in the title demonstrates the book's primary concern with barriers, whereas the two preceding lexemes serve as its modifiers: *noise*, basically a noun, but used here with the adjectival meaning complementing the head noun *barrier*, and *environmental*, an adjective proper. Semantically, barriers are obstructions resembling fences or walls and implicitly protections of public health. *Environmental*, although probably referring to the noise in people's residences and their external environment, as distinct from occupational noise, might also suggest another coupling, *environmental barriers*, suggesting thus either their character or their siting. And finally, lexeme *noise* is 'the general word for any loud, unmusical, or disagreeable sound'. Its etymology is also interesting to consider: the word can be traced back to the Latin word *nausea*. (disgust, loathing, and even a feeling of sickness at the stomach – Webster 1988: 904).

2. NEOLOGISMS AND NOISE – THREATS TO LANGUAGE AND HEARING

Language is an ever-changing system – a system in spite of all the changes that daily threaten to disrupt it. Language change, it should be emphasised, takes place within certain boundaries, which guarantees the interlocutors' mutual understanding. 'As long as we talk and write among ourselves, the language we use can vary only within understandable bounds' (Algeo 1988:XVII). Therefore, 'interpretable messages' (expression taken from Hatch-Brown 1995: 338) are absolutely essential for successful communication. People, however, do not speak in the same way. Quite the opposite, there are allowable variations in the way people use the same language and they also adapt their language to the occasions for which they use it. There are subtleties in meanings or meaning variations that are conveyed and people have different options at their disposal. The resulting forms of their individual decisions are different forms for the same meaning. Meaning and word building possibilities are infinite but some outcomes seem to be more acceptable than others and they are shared by a great majority of people. People's attitudes, however, are different when the introduction of novelties is concerned. A lot of them welcome the linguistic possibilities which they use with ease. Others do not consider new linguistic expressions acceptable – some sceptical participants in the discourse take neologisms 'as a threatening corruption of the language' (Algeo 1988: XXIII). Language, according to them, should remain the same in order to serve its role and to be worthy of its position in the cultural heritage of a nation. In the same vein, noise disrupts hearing. In a modern industrial society the problem of noise in the background of any human activity, has become particularly acute. Traffic, in the streets or on the roads, is considered to be one of the primary sources of noise. A lot of people living in the noisy environment have learned to live with noise and are not affected by it. However, there are also a lot of people who seem to be annoyed by noise and who are susceptible to its damaging effects on hearing. Hearing damage is a risk that can to be avioded by building environmental noise barriers, which serve as protections against hearing damage but which also contribute to a better quality of life. There is, therefore, an obvious similarity between the threat linguistically-prone people feel when neologisms are concerned as affecting the language they use, and the threat people feel when noise is concerned as affecting their hearing.

3. PHRASES AND COMPOUNDS

New words, phrases and compounds are therefore formed on a daily basis. One of the simplest and most productive ways of word formation is derivation, when certain prefixes and suffixes are added to the base word and when the meaning of such derived words can easily be understood if the meanings of the base word and the affix(es) are known[1] Phrases and compounds are a more sophisticated way of word formation

[1] *Some derivational suffixes, -ed and –ing, for adjective formation were subject of our analysis in the 2014 Noise and Vibration Conference paper.*

which might, to a large extent, be associated with the general trends in modern thinking and practice. The general spirit of our time can't be dissociated from the Internet communication, which is, of course, neither the sole nor temporally the earliest venue of the appearance of linguistic economy, i.e. a tendency for the linguistic expression to be concise. In this vein, phrases or even clauses could be recombined so that the resulting form is much shorter, often reduced to a single word. Formerly syntagmatically related words thus stop functioning on that plane and become individual words filled with condensed meaning. When this option is not possible, words are combined in multiple ways to form new phrases and compounds. It is, therefore, in general, more difficult to understand their meaning. Their translation also stumbles on the appearance of ambiguities which are, in practice, frequently solved with the help of the technical expert when the technical fields are concerned.

This fact is corroborated by a truism: what communicators mean becomes quite obvious when communication among peers is concerned compared to translators and linguists Peers in some occupational or technical field are even superior in the formation and interpretation of the compounds pertaining to their fields as they share the basic knowledge of the field. Such discourse is also given a special significance considering the fact that it is communication aimed at supplying as much information as possible with as few words as possible. Compounds in general are a case in point. They are also part of the technical discourse which in this way becomes more formal in nature. For the same reason, their meaning seems to be cramped which is achieved by recombining the linguistic elements, sometimes even five or six of them. The more elements there are in a compound, the more difficult it is to understand. But their use is an implication of 'a respectful and sober attitude to the referent; their use to persons of high status and responsibility is decorous; and they are a device to make communication specific, precise, and clear.' (Algeo 1988: XXIII). In any case, the creation of neologisms in general, and in scientific discourse in particular, arises out of the need for them. 'It is understandable that new inventions and notions in certain subject fields need to be lexically denoted, which entails the production of new terms to be established in terminological systems' (Stojičić 2004: 35).

4. ADJECTIVAL COMPOUNDS

Compounds are known to take a lion's share in scientific discourse. Exploring the field of electrical engineering, Miloš D. Djurić has come up with the fact that two thirds of neologisms in this field belong to compounds (Đurić 2009: 337). Compounding is actually a very powerful word building process because it presents a useful way of condensing information and adding variation to the way concepts are referred to in some discourse (Hatch and Brown 1995: 191). Although they are desirable in this respect, their understanding and translating might present a demanding task.

Adjectival compounds, although compounded, simply function as adjectives: they modify the noun following them. In terms of ortography, these adjectives can contain a hyphen, they can remain separate, or they can look like a single word. Our research was undertaken with the aim of analysing adjectival compounds in environmental noise discourse. The selected adjectives (169 in number) were all hyphenated although nothing in the way of understanding would have been different if the words had one of the two remaining possibilities in ortographic representation. In general, two groups of compound adjectives whose source were verbs were identified: one which was basically a past participle formation (*-ed*) and the other which contained the present participle (*-ing*). However, a certain number of compound adjectives containing adjective proper were identified (e.g. noise-absorptive panels). And finally, certain combinations of nouns were used to function as adjectives and modify another noun.

a) Past-Participle Compounds

Out of 48 (28.4%) compound adjectives belonging to this group, 18 or 37.5 % were formed by using the word (morpheme) *well-* as a sign of the speaker's approval or positive attitude to the quality described by the past participle adjective. Some of them were chosen to illustrate the point:

- well-designed barriers/ structures/ planting
- well-maintained environments/ grass slopes
- well-integrated barrier
- well-chosen colours and materials.

Among this group of adjectives, those denoting the shape and colour of the nouns they preceded were dominant, which is quite understandable if you know how important these considerations are in the design of environmental noise barriers:

- T-shaped barriers................ light-coloured materials
- Y-shaped barriers................ blue-coloured acrylic panels
- V-shaped concrete barrier.... dark-coloured profiled concrete panels
- mushroom-shaped forms
- tree-shaped acrylic windows

Two compond adjectives crucial for noise discourse were also identified in the book:

- A-weighted sound pressure levels
- noise-induced communication disturbance.

b) Present-Participle Compounds

Another group of compound adjectives (29 or 17.2 %) which emphasise their verbal origin and character contain the present participle:

- noise-absorbing surfaces/ materials
- sound-absorbing surfaces/ materials/ mechanisms/ panels
- load-bearing structures
- natural-looking barriers.

c) Compounds With Adjective Proper

These adjectives (20 or 11.8 %) are not derivations from the verbs, like the adjectives in the preceding two groups, they are rather adjectives in their own right, althogh the process of derivation is certainly implied in their formation:

- two-dimensional model
- sound-reflective fences
- site-specific problems.

d) Compounds With Nouns Functioning Adjectivally

As expected, this group is the largest one (72 compounds or 42.6%). Nouns preceding other nouns and acting as their modifiers, like *night-time noise levels/limits*, are quite numerous in English. The nouns serving as adjectives here are often preceded by a number or another adjective:

- two-lane roads
- long-term effects/ exposure
- high-tech image /appearance
- high-quality design.

The ratio of these four groups of compound adjectives indentified in the book could be represented in the chart in Fig.1:

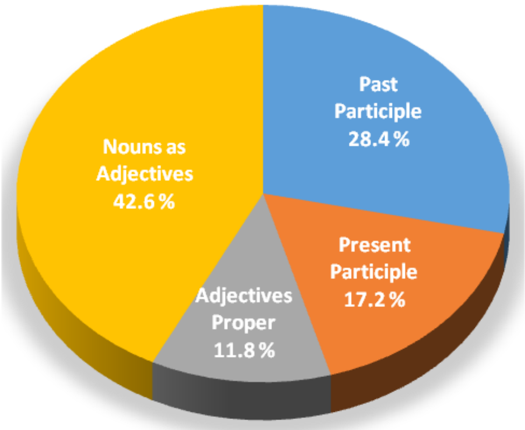

Fig.1 *The ratio of the four groups of compound adjective*

Most of the forms used as compound adjectives in the analysed book are obviously derived from verbs or belong to another class (nouns) functioning adjectivelly (88.2 %). It seems that the degree of the speaker's creativity or ingenuinity is higher if they are used than when compound adjectives serve as adjectives (11.8 %).

CONCLUSION

Complex or compound adjectives reflect the speaker's intention to make concise statements by combining more elements than is usual. Scientific discourse is particularly characterised by this productive compounding. The corpus consisting of one book only, which was subject of our analysis, is hardly sufficient for drawing general conclusions. However, it could be a contribution to some larger-scale analysis along these lines.

Adjectival compounds are forms of succint expression of complex meanings. However, it is necessary to bear in mind that languages differ in terms of their possibilities when compounding is concerned. English, already an international language, readily combines free morphemes to come to a complex meaning. Serbian, typologically different, is less likely to make compounds. Considering adjectival compounds, they appear twice as much in English than in Serbian.[2] Their translation from English into Serbian will therefore be characterised by different kinds of wording compared to English. One of much clearer forms that helps in this case is combining words into syntactic forms, most likely relative clauses. Environmental noise discourse in English will therefore have much more adjectival compounds than Serbian equivalent.

REFERENCES

[1] Algeo, John. 1988. The English Language: Variation, The Dictionary and the User in *Webster's New World Dictionary*. Cleveland: Simon and Schuster, Inc., XVII-XXIV.

[2] Đurić, Miloš D. 2009. Ka objedinjenom tumačenju identiteta složenica u diskursu elektrotehnike. In *Jezik, književnost, identitet – jezička istraživanja*, Zbornik radova, Mišić Ilić, B. and V. Lopičić (ur.) Univerzitet u Nišu, Filozofski fakultet, 336-343.

[3] Hatch, Evelyn and C. Brown. 1995. *Vocabulary, Semantics, and Language Education*. Cambridge: Cambridge University Press.

[4] Jovanović, Vladimir Ž. 2003. Frekvencijski indeks adjektivalnih složenica u engleskom i srpskom. *Primenjena lingvistika*, br. 4. Novi Sad: Jugoslovensko društvo za primenjenu lingvistiku – Filozofski fakultet u Novom Sadu. 2003. Točanac, D. I M. Jovanović (prir.), 40-48.

[5] Kotzen, Benz and C. English. 1999. *Environmental Noise Barriers. A Guide to their Acoustic and Visual Design.* London: E & FN SPON.

[6] Stojičić, Violeta. 2004. Sociolinguistic Stimuli to Development of the English Lexicon – Language Contact and Social Need. *Facta Universitatis, Series: Linguistics and Literature*, Vol. 3, No 1, 2004, pp. 29-36.

[7] Tošić, Jelica. 2014. Suffixation as a Word Formation Process. *24th International Conference Noise and Vibration, Niš 2014*, University of Niš – Faculty of Occupational Safety and 'Politechnica' University of Timisoara – Faculty of Mechanical Engineering, pp. 185-187.

[2] *Findings of the research reported by Jovanović 2003.*

2nd SESSION

VIBRATION

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THE OPTIMAL HOMOTOPY ASYMPTOTIC METHOD FOR A SYSTEM WITH LINEAR AND NONLINEAR SPRING IN SERIES

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Abstract - *In the present paper, the Optimal Homotopy Asymptotic Method (OHAM) is applied to solve large amplitude free vibration of a mass grounded by linear and nonlinear spring in series. The oscillating system is modelled by a nonlinear ordinary differential equation having linear and nonlinear stiffness components. The main objective of this study is to obtain highly accurate analytical solutions walid for the whole solution domain for free vibration of an oscillator with inertia and cubic nonlinearities. Some numerical results are presented to validate the present analysis.*

1. INTRODUCTION

In the theory of nonlinear oscillations, most attention is concentrated on the development of approximate analytical methods and their apploications to specific models. In science and enginering there exist many nonlinear and even strongly nonlinear problems which are still very difficult to solve. In some cases, inhenrent difficulties are overcome by replacing a nonlinear differential equation with a corresponding linear differential equation that approximates the original one close enough to give useful results, but this is only a harsh approximation.

To date, perturbation methods are well established tools to study diverse aspects of nonlinear problems [1], [2]. However, the use of perturbation theory in many important practical problems is invalid, or it simply breaks down for parameters beyond a certain specificed range. Therefore, new analytical techniques should be developed to overcome these shortcomings. Some extensions of the Lindstedt-Poincare perturbation method to strongly nonlinear systems have been proposed [3]. In [4] a new parameter was introduced so that it remains small regardless of the magnitude of the original parameter. In this way, a strongly nonlinear system with a large parameter is transformed into a small parameter system with respect to the new parameter.

In order to overcome the limitations related to small parameters, several effective methods were developed, such as the method of Lie group [5], the Adomian decomposition method [6], the weighted linearization method [7], the optimal variational method [8], the optimal parametric iteration method [9] and so on.

The Optimal Homotopy Asymptotic Method (OHAM) has recently became an efficient analytical technique in solving nonlinear problems. This method leads to higly accurate solutions as compared with other analytical methods. In this study, the OHAM is used to find analytical solutions for nonlinear free oscillations of a mass grounded by linear and nonlinear springs. It is shown that the solutions are quickly convergent and their components can be simply computed. The results obtained by OHAM are compared with numerical integration results and it can be observed that OHAM produces accurate results with small computational effort. The excellent accuracy of our procedure indicates that this approach can be used for solving problems where strong nonlinearities are taken into account.

2. GOVERNING EQUATION OF MOTION

In this work we will consider the system shown in fig.1 where k_1 and k_2 describe the parameters of linear spring and hardening/softening cubic nonlinear characteistic.

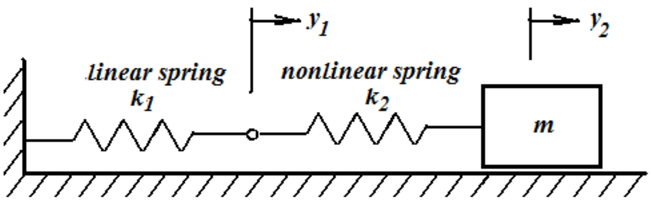

Fig. 1 *The model of a mass with serial linear and nonlinear stiffness on a frictionless contact surface*

As it is shown in fig.1, x is the net deflection of nonlinear spring defined as

$$x = y_2 - y_1 \tag{1}$$

and the force acting upon it:

$$F_2 = k_2 x + \varepsilon k_2 x^3 \tag{2}$$

where k_2 is the coefficient associated with the linear while εk_2 is the coefficient associated with the nonlinear portions of a spring force, ε is a parameter.

The equation of motion of the system presented in fig.1 can be obtained as

$$k_1 y_1 - k_2 (y_2 - y_1) - \varepsilon k_2 (y_2 - y_1)^3 = 0 \tag{3}$$

$$m\ddot{y}_2 + k_2 (y_2 - y_1) + \varepsilon k_2 (y_2 - y_1)^3 = 0 \tag{4}$$

Form eqs. (1) and (2) it holds that

$$y_1 = \frac{k_2}{k_1}x + \varepsilon\frac{k_2}{k_1}x^3 \quad (5)$$

By differentiating twice eq. (5) with respect to time and substituting it into eq.(4), it can be shown that

$$m\left(1+\frac{k_2}{k_1}+3\varepsilon\frac{k_2}{k_1}x^2\right)\ddot{x}+6m\varepsilon\frac{k_2}{k_1}x\dot{x}^2+k_2x+\varepsilon k_2x^3=0 \quad (6)$$

The eq.(6) retrieves

$$(1+\alpha x^2)\ddot{x}+2\alpha x\dot{x}^2+\beta x+\varepsilon\beta x^3=0 \quad (7)$$

where

$$\alpha=\frac{3\varepsilon k_2}{m(k_1+k_2)}, \beta=\frac{k_1k_2}{m(k_1+k_2)} \quad (8)$$

It should be emphasized that a term proportional to velocity squared appears into eq.(7), suggesting that the system contains a dissipative element although this is not the case [12]. The analytical solution to be presented in what follows will be in terms of x. Therefore, it seems meaningful to give the initial condition in x:

$$x(0)=A, \dot{x}(0)=0 \quad (9)$$

If we make the transformation

$$\tau=\Omega t, x(t)=Ay(\tau) \quad (10)$$

then the eqs.(7) and (9) can be written in the forms

$$\begin{aligned}&(1+\alpha A^2y^2)y''+2\alpha A^2yy'^2+\beta\Omega^{-2}y+\varepsilon\beta\Omega^{-2}A^2y^3=0\\&y(0)=1, y'(0)=0\end{aligned} \quad (11)$$

where the prime denotes derivative with respect to τ.

3. BASIC CONCEPTS OF OHAM [10], [11]

To apply OHAM in the second alternative, we consider the nonlinear differential equation

$$L[(y(\tau)]+g(\tau)+N[y(\tau)]=0, \tau\in D \quad (12)$$

subject to the initial conditions

$$B\left(y(\tau),\frac{dy(\tau)}{d\tau}\right)=0 \quad (13)$$

where L is a linear operator, τ denotes independent variable, y(τ) is an unknown function, g(τ) is a known function, N[y(τ)] is a nonlinear operator, D is the domain of interest and B is a boundary operator.

By means of OHAM, one first construct a family of equations:

$$\begin{aligned}&(1-p)[L(\tau,p,C_i)+g(\tau)]=\\&=H(\tau,p,C_i)[L(\tau,p,C_i)+g(\tau)+N(y(\tau,p,C_i))]\end{aligned} \quad (14)$$

where $p\in[0,1]$ is an embedding parameter, $H(\tau,p,C_i)$ is a nonzero auxiliary function for $p\neq 0$ and $H(\tau,0,C_i)=0$, $y(\tau,p,C_i)$ is an unknown function, C_i are unknown parameters which will be determined later.

Obviously, when p=0 and p=1, it holds

$$y(\tau,0,C_i)=y_0(\tau) \quad (15)$$

$$y(\tau,1,C_i)=y(\tau,C_i) \quad (16)$$

Therefore, as p increases from 0 to 1, the solution $y(\tau,p,C_i)$ varies from $y_0(\tau)$ to the solution $y(\tau,C_i)$, where the initial approximation $y_0(\tau)$ is obtained from eq.(14) for p=0:

$$L[y_0(\tau)]+g(\tau)=0 \quad (17)$$

and

$$B\left(y_0(\tau),\frac{dy_0}{d\tau}\right)=0 \quad (18)$$

After only one iteration we obtain

$$y(\tau,p,C_i)=y_0(\tau)+py_1(\tau,C_i) \quad (19)$$

such that the first-order approximate solution of eq.(12) is given by

$$\bar{y}(\tau,C_i)=y(\tau,1,C_i)=y_0(\tau)+y_1(\tau,C_i) \quad (20)$$

where the first approximation $y_1(\tau,C_i)$ is obtained from eqs.(19) and (14), equating the coefficients of p:

$$L[y_1(\tau,C_i)]=H(\tau,C_i)N[y_0(\tau)] \quad (21)$$

The initial conditions are

$$B\left(y_1(\tau,C_i),\frac{dy_1(\tau,C_i)}{d\tau}\right)=0 \quad (22)$$

We remark that $y_0(\tau)$ and $y_1(\tau,C_i)$ are obtained from the linear differential equations (17) and (21), with the initial conditions given by eqs. (18) and (22) respectively. The convergence of the approximate solutions (20) depends upon the auxiliary function $H(\tau,C_i)$. There are many possibilities to choose the function $H(\tau,C_i)$. The convergence of the solution $y_1(\tau,C_i)$ and consequently the convergence of the approximate solution $\bar{y}(\tau,C_i)$ depend on the auxiliaty function $H(\tau,C_i)$. Basically, the shape of $H(\tau,C_i)$ must follow the terms appearing in eq.(21). Therefore, we try to choose $H(\tau,C_i)$ so that in eq.(21), the product $H(\tau,C_i)N[y_0(\tau)]$ be of the same shape with the terms which appear into $N[y_0(\tau)]$.

The first-order approximate solution (20) depends on the convergence-control parameters C_i, i=1,2,....,s, The values of these parameters can be optimally identified via various methods, such as the least square method, the Galerkin method, the collocation method, the Ritz method and so on.

It would be preferred to use

$$J(C_1,C_2,...,C_s)=\int_a^b R^2(\tau,C_1,C_2,...,C_s)d\tau \quad (23)$$

where a and b are values depending on the given problem and R is given by $R(\tau,C_i)=L[\bar{y}(\tau,C_i)+g(\tau)+N(\bar{y}(\tau,C_i))]$. The optimal values of the convergence-control parameters C_1, C_2, ...,C_s can be identified from the conditions

$$\frac{\partial J}{\partial C_1}=\frac{\partial J}{\partial C_2}=...=\frac{\partial J}{\partial C_s} \quad (24)$$

After obtaining these optimal values of these convergence-control parameters, the first-order approximate solution given by eq.(20) is well-determined.

4. APPROXIMATE SOLUTION OF EQ.(11) BY OHAM

The eq.(11) can be rewritten in the form

$$y'' + y + \alpha A^2 y^2 y'' + 2\alpha A^2 yy'^2 + (\beta\Omega^{-2} - 1)y + \\ + \varepsilon\beta\Omega^{-2} A^2 y^3 = 0, y(0) = 1, y'(0) = 0 \quad (25)$$

Consequently, the linear and nonlinear operator are given from eq.(25) (g(τ)=0):

$$L[y(\tau)] = y'' + y \quad (26)$$

$$N[y(\tau,\Omega)] = \alpha A^2 y^2 y'' + 2\alpha A^2 yy'^2 + \\ + (\beta\Omega^{-2} - 1)y + \varepsilon\beta\Omega^{-2} A^2 y^3 \quad (27)$$

The eq.(17) becomes

$$y_0'' + y_0 = 0, y_0(0) = 1, y'(0) = 0 \quad (28)$$

whose solution is

$$y_0(\tau) = \cos\tau \quad (29)$$

Substituting eq.(29) into eq.(27) we have

$$N[y_0(\tau),\Omega] = M\cos\tau + N\cos 3\tau \quad (30)$$

where

$$M = \frac{A^2}{4}(3\varepsilon\beta\Omega^{-2} - \alpha) + \beta\Omega^{-2} - 1 \quad (31)$$

$$N = \frac{A^2}{4}(\varepsilon\beta\Omega^{-2} - 3\alpha) \quad (32)$$

The differential equation corresponding to the first approximation is obtained from eq.(21)

$$y_1'' + y_1 = H(\tau, C_i)(M\cos\tau + N\cos 3\tau), y_1(0) = y_1'(0) = 0 \quad (33)$$

If we choose the auxiliary function as

$$H(\tau, C_i) = C_1 + 2C_2\cos 2\tau + 2C_3\cos 4\tau \quad (34)$$

then substituting eq.(34) into eq.(33) it holds that

$$y_1'' + y_1 = (MC_1 + MC_2 + NC_2 + NC_3)\cos\tau + \\ (MC_2 + MC_3 + NC_1)\cos 3\tau + (MC_3 + \\ + NC_2)\cos 5\tau + NC_3\cos 7\tau \quad (35)$$

Now, avoiding the presence of a secular term into eq.(35) needs

$$\Omega^2 = \frac{(3\varepsilon\beta A^2 + 4\beta)C_1 + 4(\varepsilon\beta A^2 + \beta)C_2 + \varepsilon\beta A^2 C_3}{(\alpha A^2 + 4)C_1 + 4(\alpha A^2 + 1)C_2 + 3\alpha A^2 C_3} \quad (36)$$

Form eq.(33) it holds that

$$y_1(\tau, C_i) = \frac{NC_1 + MC_2 + MC_3}{8}(\cos\tau - \cos 3\tau) + \\ \frac{NC_2 + MC_3}{24}(\cos\tau - \cos 5\tau) + \frac{NC_3}{48}(\cos\tau - \cos 7\tau) \quad (37)$$

From eqs.(29), (37) , (20 and (10) one retrieves the first-order approximate solution of eqs. (7) and (9):

$$\bar{x}(t, C_i) = A\cos\Omega t + \\ + \frac{A(NC_1 + MC_2 + MC_3)}{8}(\cos\Omega t - \cos 3\Omega t) + \\ \frac{A(NC_2 + MC_3)}{24}(\cos\Omega t - \cos 5\Omega t) + \\ + \frac{ANC_3}{48}(\cos\Omega t - \cos 7\Omega t) \quad (38)$$

where Ω is obtained from eq. (36).

5. NUMERICAL EXAMPLES

We illustrate the accuracy of our procedure for some values of the parameters α, β, ε and A, comparing the obtained analytical solutions with the numerical integration results obtained using a fourth-order Runge Kutta method.

5.1 Case 1

For α=0.4, β=0.5, ε=0.6, A=1, we obtain the optimal values of the convergence-control parameters and the frequency as C_1=-0.862135448, C_2=0.153375876, C_3=-0.107690942 and Ω=0.81634, and consequently, the first-order approximate periodic solution (38) becomes

$$\bar{x}(t) = 1.01941\cos\Omega t - 0.0201326\cos 3\Omega t + \\ + 0.00114376\cos 5\Omega t - 0.000420571\cos 7\Omega t \quad (39)$$

5.2 Case 2

For α=0.3, β=0.4, ε=0.7,A=0.5, we obtain C_1=-0.946732473, C_2=0.087944355, C_3=0.031002724 and Ω=0.66723. In this case the first-order approximate solution becomes

$$\bar{x}(t) = 0.500947\cos\Omega t - 0.000985002\cos 3\Omega t + \\ + 0.0000325555\cos 5\Omega t + 5.4712 * 10^{-6}\cos 7\Omega t \quad (40)$$

Figures 2 and 3 present a comparison between the analytical solutions (39) and (40) and numerical results in the considered cases.

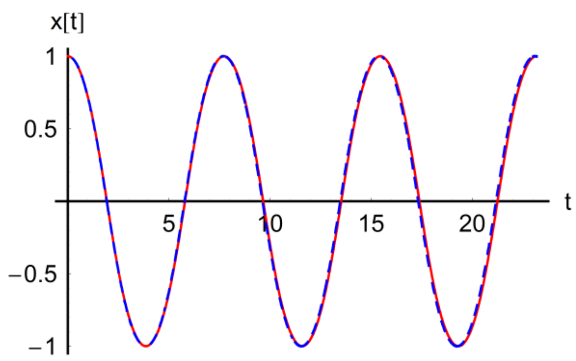

Fig. 2 *Comparison of the results in case 1: α=0.4, β=0.5, ε=0.6, A=1_____ numerical results; _ _ _ _ _ OHAM solution (39)*

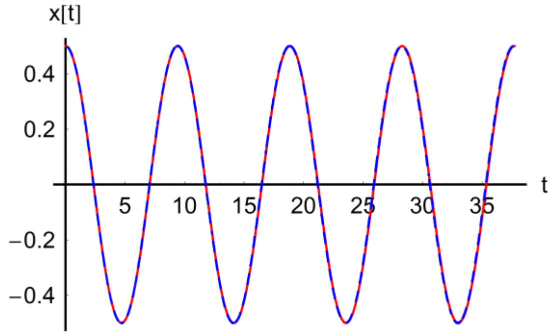

Fig. 3 *Comparison of the results in case 2: α=0.3, β=0.4, ε=0.7, A=0.5_____ numerical results; _ _ _ _ _ OHAM solution (40)*

It can be seen from the above figures that the solutions obtained by the proposed procedure are nearly identical with the solution obtaiuned by numerical integration. We also remark from Table 1 a very good agreement between the numerical and approximate values of the frequencies.

Table 1 *Comparison between the frequencies*

Case	$\Omega_{numerical}$	$\Omega_{approximate}$
5.1	0.81352	0.81634
5.2	0.66652	0.66723

6. CONCLUSIONS

In this work, the Optimal Homotopy Asymptotic Method (OHAM) is employed to propose an analytic approximate solution to nonlinear oscillations of a mass grounded by linear and nonlinear springs. Our procedure is vaklid even if the nonlinear equation does not contain any small parameters and very good approximations are obtained in few terms. It should be emphasized that our procedure converges to the exact solutions after only one iteration and this fact proves that this approach is very efficient in practice.

REFERENCES

[1] J.D. Cole, *Perturbation Methods in Applied Mathematics*, Blaudel, Waltham, 1968.

[2] A.H. Nayfeh, D.T. Mook, *Nonlinear Oscillations*, New York: Wiley, 1979

[3] Y.K. Cheung, S.H. Chen, S.L. Lau, "A modified Lindstedt-Poincare method for certain strongly non-linear oscillators", *Int. J. Non-Linear Mech.*, vol.26, pp. 367-378, 1991

[4] M. Senator, C.N. Bayat, "A perturbation technique that works even when the nonlinearity is not small", *J. Sound Vibr.*, vol. 104, pp.1-10, 1993

[5] P.J. Olver, *Application of Lie Group to Differential Equations*, New York: Springer, 1986

[6] G. Adomain, "A review of the decomposition method in applied mathematics", J. Math. Anal. Appl. Vol. 135, pp.501-541, 1988

[7] V.P. Agrawal, N.N. Denman, "Weighted linearization technique for period approximation in large amplitude nonlinear oscillators", *J. Sound Vibr.*, vol.99, pp.463-473, 1985

[8] N. Herisanu, V. Marinca, "An optimal approach to study the nonlinear behaviour of a rotating electrical machine, *J. Appl. Math.*, vol.2012, Art. ID 465023

[9] V. Marinca, N. Herisanu, *Nonlinear Dynamical Systems in Engineering. Some Approximate Approaches*, Berlin: Springer, 2011

[10] V. Marinca, N. Herisanu, "On the flow of a Walter type B' viscoelastic fluid in a vertical channel with porous wall, *Int. J. Heat and Mass Transf.*, vol.79, pp.145-165, 2014

[11] V. Marinca, N. Herisanu, *The Optimal Homotopy Asymptotic Method. Engineering Applications*, Cham: Springer, 2015

[12] M. Bayat, M. Bayat, M. Bayat, An analytical approach on a mass grounded by linear and nonlinear springs in series, *Int. J. Phys. Sci.*, vol. 6, pp.229-236, 2011

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VIBRATION OF THE CARBON NANOTUBE WITH FLUID FLOW

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Abstract - *The nonlinear vibration of an embedded single-walled carbon nanotube conveying fluid is investigated by means of the optimal vatiational method (OVM). The nonlocal continuum theory is utilized to simulate the simple-walled nanotubes (SWNTs) employing Pasternak-type elastic foundation*

1. INTRODUCTION

The discovery of carbon nanotubes (CNT) by Iijima [1] in 1991, especially the discovery of the single-wall nanotube and successful composition of the CNT in the macrography scale, has received considerable attention in recent years.

Nanotechnology is an industrial revolution and one of the most promising researches in the field of mechanics, physics, chemistry, material science and so on. A wide range of applications of CNT have been reported in the literature, including applications in nanoelectronics, nanodevices and nanocomposites [2-4].

Based on the non-coaxial vibrational model of multi-wall nanotubes (MWCNT), Ru [5] studied the buckling of nanotubes considering the intertube radial displacement and the internal Waals force, Yoon et al. [6] analyzed the frequencies and modes of the linear free vibration for the embedded multiwall nanotube. An interesting review on the vibration of CNT is given by Gibson et al. [7] including a concise review of many of the relevant publications as possible. Based on the theory of thermal elasticity mechanics, Wang et al. [8] studied the vibration and instability analysis of fluid-conveying SWNT considering the thermal effect.

However, most of the investigations conducted on the vibration of CNTs have been restricted to the linear regime and fewer works were done on the nonlinear vibration of these materials. Fu et al. [9] studied the nonlinear vibrations of embedded nanotubes using the incremental harmonic balanced method. A SWNT and double-walled nanotubes are considered in this work. The composite nanofibers with various MWCNT contents were fabricated by electrospining process and their microwave absorption. Properties were evaluated in the frequency range of 8-12 GHz at room temperature [10].

Pantano et al. [11] investigated the effect of the characteristic wave or wrinkles on the bending mode of CNT under considering the geometric nonlinearity and explained the phenomenon that the curve modes of CNT decrease with the increase of the diameter of CNT. Nonlocal discrete and continous models were developed for vibration analysis of two-dimensional ensembles of SWCNTs subjected to laterally applied loads by Kiami [12]. Transverse wave characterization within 3D ensembles of SWCNTs was aimed to be carefully studied by Kiami [13]. He used nonloncal continuum theory of Eringen and Hamilton’s principle to develop the nonlocal-discrete equation of motion of the problem based on the Rayleigh, Timoshenko and higher-order theories.

Mathematical modelling is a vantage point to reach a solution in an engineering problem, so the accurate modelling of nonlinear engineering problems is an important step to obtain accurate solutions. Most of nonlinear differential equations related to engineering problems do not have exact analytic solutions so approximations and numerical methods must be used. Different methods have been introduced to solve these equations such as the averaging methods [14], the Lindstedt-Poincare method [15], a modified Lindstedt-Poincare method [16], the method of harmonic balance [15], [17], the weighted linearization method [18], the variational iteration method [19], the optimal iteration method [20], the optimal homotopy perturbation method [21] and so on [22], [23].

All the above mentioned methods work very well for weakly nonlinear dynamical systems and some of then work even for strongly nonlinear problems. It is very important in case of strongly nonlinear dynamical systems to prove and to ensure the conditions of convergence of the solutions and every approach used in the study of nonlinear dynamical systems must be rigorously proved.

In the present work, the nonlocal continuum theory is utilized to simulate the nonlinear vibrations of a SWCNT conveying fluid, employing Pasternak-type elastic foundation. To solve the governing equations of the problem, one of the strongest approximate method, namely the optimal variational method is used. Our procedure does not depend upon any small or large parameters, contradistinguishing from other known methods. The main advantage of this approach is the control of the convergence of approximate solutions in a very rigorous way. A very good agreement was found between the approximate solutions and numerical ones, which proves that our method is very efficient and accurate.

2. GOVERNING EQUATION OF MOTION

A single-walled carbon nanotube with fluid flow embedded in elastic medium is assumed to be simply supported at both ends and the effect of gravity is negligible. For Bernouli-

Euler beam theory, the relationshipbetween the transverse shear force Q, the bending moment M and the longitudinal force N is [24], [25]:

$$\frac{\partial Q}{\partial x} = \frac{\partial^2 M}{\partial x^2} + N\frac{\partial^2 W}{\partial x^2} \tag{1}$$

$$\begin{aligned} M &= \int z\delta_{xx} dA_c = \int zE\varepsilon_{xx} dA_c \\ N &= \int \sigma_{xx} dA_c = \int E\varepsilon_{xx} dA_c \end{aligned} \tag{2}$$

where σ_{xx} is the nonlocal stress component in the x direction, E is the modulus of elasticity, A_c is the cross-sectional area and ε_{xx} is the Karman strain. The following relations can be used

$$\int z dA_c = 0, \qquad \int z^2 dA_c = I \tag{3}$$

The nonlocal constitutive equation for the uniaxial bending stress state forms as

$$\sigma_{xx} = E\varepsilon_{xx} + (e_0 a)^2 \frac{\partial^2 \sigma_{xx}}{\partial x^2} \tag{4}$$

where e_0a is a nonlocal parameter, showing the small scale effect. The nonlocal continuum theory presented by Eringen [26], shows a more precise constitutive rule for small-scale structures in comparison with the common local elastic theories, such that

$$M - (e_0 a)^2 \frac{\partial^2 M}{\partial x^2} = \int zE\varepsilon_{xx} dA_c \tag{5}$$

Based on the Bernouli-Euler continuum theory, the displacement field of the model is expressed as

$$U(x,z,t) = u(x,t) - z\frac{\partial W}{\partial x}, \qquad W(x,z,t) = W(x,t) \tag{6}$$

Also, the von-Karman strain is given by

$$\begin{aligned} \varepsilon_{xx} &= \frac{\partial U(x,z,t)}{\partial x} + \frac{1}{2}\left(\frac{\partial W(x,z,t)}{\partial x}\right)^2 = \\ &= \frac{\partial^2 U}{\partial x^2} - z\frac{\partial^2 W}{\partial x^2} + \frac{1}{2}\left(\frac{\partial W(x,t)}{\partial x}\right)^2 \end{aligned} \tag{7}$$

From Eqs. (4) and (7), the nonlocal stress resultant can be defined as

$$M - (e_0 a)^2 \frac{\partial^2 M}{\partial x^2} = EI\frac{\partial^2 W}{\partial x^2} \tag{8}$$

From Eqs. (1) and (8) we obtain

$$M = (e_0 a)^2 \left[\frac{\partial Q}{\partial x} - N\frac{\partial^2 W}{\partial x^2}\right] + EI\frac{\partial^2 W}{\partial x^2} \tag{9}$$

The governing equation of motion is obtained from Eqs. (8) and (9) as

$$\begin{aligned} &EI\frac{\partial^4 W}{\partial x^4} + \frac{dQ}{dx} - N\frac{\partial^2 W}{\partial x^2} - \\ &-(e_0 a)^2\left[\frac{\partial^3 Q}{\partial x^3} - N\frac{\partial^4 W}{\partial x^4}\right] = 0 \end{aligned} \tag{10}$$

Therefore, the governing equations for fluid-conveying SWCNT can be written as

$$\begin{aligned} &m_c\frac{\partial^2 W}{\partial t^2} + EI\frac{\partial^4 W}{\partial x^4} + k_0 W - k_p\frac{\partial^2 W}{\partial x^2} + F\frac{\partial^2 W}{\partial x^2} + \\ &+ m_f\left(2\nu\frac{\partial^2 W}{\partial x \partial t} + \nu^2\frac{\partial^2 W}{\partial x^2} + \frac{\partial^2 W}{\partial t^2}\right) - \\ &-(e_0 a)^2\left[m_c\frac{\partial^4 W}{\partial t^2 \partial x^2} + k_e\frac{\partial^2 W}{\partial x^2} - k_p\frac{\partial^4 W}{\partial x^4} + \right. \\ &+ F\frac{\partial^4 W}{\partial x^4} + m_f\left(2\nu\frac{\partial^4 W}{\partial x^3 \partial t} + \nu^2\frac{\partial^4 W}{\partial x^4} + \frac{\partial^4 W}{\partial x^2 \partial t^2}\right) - \\ &\left. -\frac{EA_c}{2l}\frac{\partial^4 W}{\partial x^4}\int_0^l \left(\frac{\partial W}{\partial x}\right)^2 dx\right] - \frac{EA_c}{2l}\frac{\partial^2 W}{\partial x^2}\int_0^l \left(\frac{\partial W}{\partial x}\right)^2 dx = 0 \end{aligned} \tag{11}$$

The deflection of nanotube is subjected to the following conditions

$$\begin{aligned} W(0,t) &= \frac{\partial^2 W(0,t)}{\partial x^2} = 0 \qquad at \quad x = 0 \\ W(L,t) &= \frac{\partial^2 W(L,t)}{\partial x^2} = 0 \qquad at \quad x = l \end{aligned} \tag{12}$$

and W(x,t) can be expanded as

$$W(x,t) = \varphi(x) q(t) \tag{13}$$

The function $\varphi(x)$ is the normalized mode function of the nanotube from the linear vibration analysis due to the specified boundary conditions.

The beam is supposed to be simply supported at the ends, so that for the boundary conditions we have

$$\varphi(x) = \sqrt{2}\sin\frac{\pi x}{l} \tag{14}$$

Substitution of Eq.(13) and (14) into Eq.(11) leads to

$$\ddot{q} + aq + bq^3 = 0, \qquad q(0) = A, \dot{q}(0) = 0 \tag{15}$$

where

$$\begin{aligned} &e = \frac{\pi}{l}(e_0 a), \quad K_c = \frac{l^4}{\pi^4}\frac{K_c}{EI}, \; K_p = \frac{l^2}{\pi^2}\frac{K_p}{EI} \\ &\omega_0 = \frac{\pi^2}{l^2}\sqrt{\frac{EI}{m_c + m_f}}, \; T = \frac{l^2}{\pi^2}\frac{F}{EI}, \; U = \frac{l}{\pi}\sqrt{\frac{m_f}{EI}}, \\ &a = \frac{1 + e^2(K_c + K_p - T - U^2) + K_c + K_p - T - U^2}{1 + e^2}\omega_0^2 \\ &r = \sqrt{\frac{I}{A_c}}, \; b = \frac{\omega_0^2}{4r^2} \end{aligned} \tag{16}$$

3. BASIC OF THE PROPOSED METHOD

In order to show the basics of the OVM, we consider the following differential equation

$$F(\tau, q, q', q'') = 0 \tag{17}$$

The variational principle can be easely established if there exists a functional

$$J = \int_0^{T/4} L(\tau, q, q')d\tau \tag{18}$$

which admits as extremals the solutions of Eq. (17), where L is the Lagrangian of the system and T is the period.

This problem is based on the study of the conditions under which there exists a function L(τ,q,q') such that Euler's equation of the functional (17) coincide with the system (17) that is

$$\frac{\partial L}{\partial q} - \frac{d}{d\tau}\left(\frac{\partial L}{\partial q'}\right) = F(\tau, q, q', q'') \tag{19}$$

In our procedure, we assume that the approximate solution $\bar{q}$ depends on several parameters C_1, C_2,...C_s

$$\bar{q} = \bar{q}(\tau, C_1, C_2, ..., C_s) \tag{20}$$

such that the action functional (18) becomes

$$J(C_1, C_2, ..., C_s, \Omega) = \int_0^{T/4} L(\tau, \bar{q}(\tau, C_i), \bar{q}'(\tau, C_i))d\tau \qquad i = 1,2,...,s \tag{21}$$

The parameters C_i which appear in Eq. (21), called convergence-control parameters, can be determined optimally applying the Ritz method [22], [23]:

$$\frac{\partial J}{\partial C_1} = \frac{\partial J}{\partial C_2} = ... = \frac{\partial J}{\partial Cs} \tag{22}$$

From Eq. (21) and from the initial conditions which becomes

$$\bar{q}(0, C_1, C_2, ..., C_s) = 1 \tag{23}$$

we can obtain the optimal values of the convergence-control parameters C_1, C_2,...,C_s and the frequency Ω of the system (17).

We remark that the expression (20) of the solution is not unique.

4. APPLICATION OF THE VIBRATION OF CARBON NANOTUBE WITH FLUID FLOW

The validity of our procedure is illustrated in what follows. In the analysed case the Lagrangian cam be written as

$$L(\tau, q, q') = -\frac{1}{2}\Omega^2 q'^2 + \frac{1}{2}aq^2 + \frac{1}{4}bA^2 q^4 \tag{24}$$

If we consider s=3 in Eq. (20), then the approximate solution of the considered equation can be written as

$$\bar{q}(\tau) = C_1 \cos\tau + C_2 \cos 3\tau + C_3 \cos 5\tau \tag{25}$$

Also, we can choose this approximate solution in other forms, such as

$$\bar{q}(\tau) = C_1 \cos\tau + C_2 \cos 5\tau + C_3 \cos 7\tau \tag{26}$$

or

$$\bar{q}(\tau) = C_1 \cos\tau + C_2 \cos 3\tau + C_3 \cos 7\tau \tag{27}$$

and so on.

Substituting Eq. (25) into (24) and this into Eq. (21), we have the following result:

$$\begin{aligned} J(C_1, C_2, C_3, \Omega) &= -\frac{1}{4}\Omega^2(C_1^2 + 9C_2^2 + 25C_3^2) + \\ &+ \frac{1}{4}a(C_1^2 + C_2^2 + C_3^2) + \frac{1}{4}bA^2\left(\frac{3}{8}C_1^4 + \frac{3}{8}C_2^4 + \right. \\ &+ \frac{3}{8}C_3^4 + \frac{3}{2}C_1^2C_2^2 + \frac{3}{2}C_1^2C_3^2 + \frac{3}{2}C_2^2C_3^2 - \\ &\left. - \frac{1}{2}C_1^3C_2 + \frac{1}{2}C_1^2C_2C_3 + \frac{1}{2}C_1C_2^2C_3\right) \end{aligned} \tag{28}$$

The values of the parameters C_1, C_2, C_3 and the frequency Ω are obtained from Eqs. (22) and (23) which become

$$\begin{aligned} &-\frac{1}{2}\Omega^2C_1 + \frac{1}{2}aC_1 + \frac{1}{4}bA^2\left(\frac{3}{2}C_1^3 + 3C_1C_2^2 + 3C_1C_3^2 - \right. \\ &\left. -\frac{3}{2}C_1^2C_2 + C_1C_2C_3 + \frac{1}{2}C_2^2C_3\right) = 0 \\ &-\frac{9}{2}\Omega^2C_2 + \frac{1}{2}aC_2 + \frac{1}{4}bA^2\left(\frac{3}{2}C_2^3 + 3C_1^2C_2 + 3C_2C_3^2 - \right. \\ &\left. -\frac{1}{2}C_1^3 + \frac{1}{2}C_1^2C_3 + C_1C_2C_3\right) = 0 \\ &-\frac{25}{2}\Omega^2C_3 + \frac{1}{2}aC_3 + \frac{1}{4}bA^2\left(\frac{3}{2}C_3^3 + 3C_1^2C_3 + 3C_2^2C_3 + \right. \\ &\left. +\frac{1}{2}C_1^2C_2 + \frac{1}{2}C_1C_2^2\right) = 0 \\ &C_1 + C_2 + C_3 = 1 \end{aligned} \tag{29}$$

5. TEST EXAMPLES

The validity of the proposed procedure for solving the investigated problem is illustrated on two examples, considering different parameters and different initial amplitudes.

5.1 Case 1

As a first example, we consider the following set of parameters: a=0.6, b=0.4, A=0.5. The optimal values of the convergence-control parameters and the frequency Ω are obtained from Eqs. (22) as

C_1=1.00559,C_2=-0.00521933,C_3=-0.000370246,Ω=0.822783

The approximate solution of the considered equation in this case becomes

$$\begin{aligned} \bar{q}(t) = A\bar{q}(\Omega t) &= 0.502795\cos\Omega t - \\ &- 0.00260967\cos 3\Omega t - 0.0000575561\cos 5\Omega t \end{aligned} \tag{30}$$

Figure 1 shows the comparison between the present solution (30) and numerical integration results obtained using a fourth-order Runge-Kutta method.

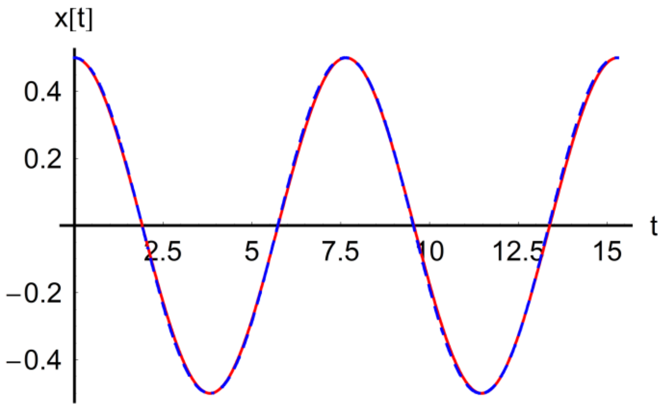

Fig. 1 *Comparison between analytical and numerical results results in case 5.1:*

_____ *numerical results;* _ _ _ _ _ *analytical solution (30)*

5.2 Case 2

As a second example we consider a=0.5, b=0.3, A=0.8. Following the same procedure, we obtain the optimal values of the convergence-control parameters and the frequency as:

C_1=0.997443, C_2=-5.99956*10^{-6}, C_3=0.00256345, Ω=0.800177

and consequently, the approximate solution will be

$$\bar{q}(t) = A\bar{q}(\Omega t) = 0.797954\cos\Omega t - \\ -4.79965\cdot 10^{-6}\cos 3\Omega t + 0.00205076\cos 5\Omega t \qquad (31)$$

The comparison between the present solution (31) and numerical integration results in this case is presented in Fig.2.

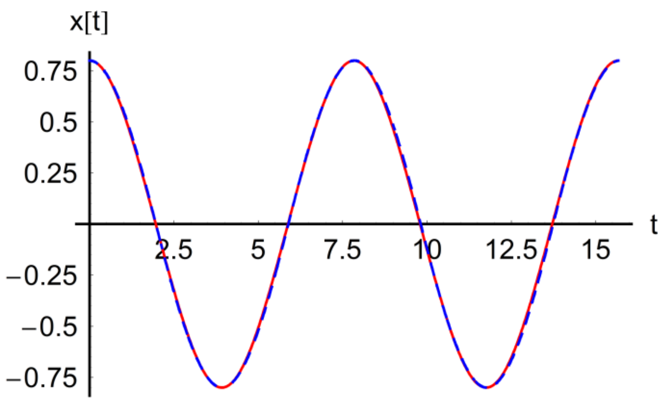

Fig. 2 *Comparison between analytical and numerical results results in case 5.2:*

_____ *numerical results;* _ _ _ _ _ *analytical solution (31)*

It can be observed from Figs. 1 and 2 that the approximate analytical results obtained using OVM are in very good agreement with the numerical ones.

6. CONCLUSIONS

In this paper, an optimal variational approach is employed to propose a new analytic approximate solution for nonlinear conservative oscillators of carbon nanotube with fluid flow. The construction of our variational approach is different from the traditional approach especially concerning the involvement of some initially unknown parameters C_i called convergence-control parameters, whose optimal values ensure a fast convergence of the approximate analytical solution.This is the main advantage of the OVM, which provides us with a simple but rigorous way to control and adjust the convergence of the solution. The proposed procedure is valid even if the nonlinear equation does not contain any small or large paratemeters. The given examples illustrate that the proposed analytical approach is very effective and yields accurate results comparing to those obtained via numerical integration.

REFERENCES

[1] S. Iijima, “Helical microtubes of graphitic carbon”, *Nature*, vol.354, pp.56-58, 1991

[2] H.Y. Song, X.W. Zha, “Mechanical properties of nickel-coated single walled carbon nanotubes and their embedded gold matrix composites”, *Phys. Lett.A*, vol.374, pp.1068-1072, 2010

[3] I. Deretzis, A. La Magna, “Electric transport in carbon nanotubes based nanodevices”, *Phys. E*, vol. 40, pp.2333-2338, 2008

[4] R. Chowdhury, S. Adhikari, J. Mitelel, “Vibrating carbon nanotube based biosensors *Phys. E*, vol. 42, pp.104-109, 2009

[5] C.Q. Ru, “Column buckling of multi-wall carbon nanotubes with interlayer radial displacements”, *Phys. Review B*, vol.62, pp.16962-16967, 2000

[6] J. Yoon, C.Q. Ru, A. Mioduchowski, “Vibration of an embedded multiwall carbon nanotube”, *Compos. Sci. Technol*., vol.63, pp.1533-1542, 2003

[7] R.F. Gibson, E.O. Ayorinde, Y Wesa, “Vibration of carbon nanotubes and their composites: a review”, *Compos. Sci. Technol.*, vol. 67, pp. 1-28, 2007

[8] L. Wang, Q. Ni, M. Li, Q. Qian, “The thermal effect on vibration and instability of carbon nanotubes conveying fluid”, *Phys. E*, vol. 40, pp. 3179-3182

[9] Y.M. Fu, J.W. Hong, X.Q. Wing, “Analysis of nonlinear vibration for embedded carbon nanotubes”, *J. Sound Vibr.*, vol. 296, pp.746-756, 2006

[10] K. Nasouri, P. Valipour, “Fabrication of polyamide carbon nanotubes composite electrospun nanofibers for microwave absorption”, *Polym. Sci. Ser. A*, vol. 57, pp. 359-364, 2015

[11] A. Pantano, M.C. Boyce, D.M. Parks, “Nonlinear structural mechanics based modelling of carbon nanotube

deformation", *Phys. Review Lett.*, vol. 91, pp.145501-1-145501-4, 2003

[12] K. Kiani, "Nonlocal continuous models for forced vibration analysis of two and three-dimensional ensembles of single-walled carbon nanotubes", *Phys.E*, vol. 60, pp.229-245, 2014

[13] K. Kiani, "Wave characteristics in aligned forests of single-walled carbon nanotubes using nonlocal discrete and continuous theories", *Int. J. Mech. Sci.*, vol.90, pp.278-309, 2015

[14] Y. Mitropolski, *The average method in nonlinear mechanics*, Kiev, Naukova Dumka, 1971

[15] A.H. Nayfeh, D.T. Mook, *Nonlinear Oscillations*, New York, Wiley, 1974

[16] S.N. Chen, Y.K. Cheung, "A modified Lindstedt-Poincare method for strongly nonlinear two degree of freedom system", *J. Sound Vibr.*, vol.193, pp.751-762, 1996

[17] P. Hagedorn, *Nonlinear Oscillations*, Oxford, Clarendon Press, 1981

[18] V.P. Agraval, N.N. Denman, "Weighted linearization technique for period approximation in large amplitude nonlinear oscillation", *J. Sound Vibr.*, vol. 99, pp. 463-473, 1985

[19] J.H. He, "Variational iteration method, a kind of nonlinear analytical technique. Some examples", *J. Non-Linear Mech.*, vol.34, pp.699-708

[20] V. Marinca, N. Herisanu, "An optimal iteration method for strongly nonlinear oscillators", *J. Appl. Math.*, ID 906341, 2012

[21] N. Herisanu, V. Marinca, Optimal homotopy perturbation method for a non-conservative dynamical system of a rotating electrical machine", *Zeit. Naturforsch. A*, vol. 67, pp.509-516, 2012

[22] V. Marinca, N. Herisanu, *Nonlinear Dynamical Systems in Engineering. Some Approximate Approaches*, Berlin, Springer, 2011

[23] V. Marinca. N. Herisanu, *The Optimal Homotopy Asymptotic Method. Engineering Applications*, Cham, Springer, 2015

[24] J. Reddy, S. Pang, "Nonlocal continuum theories of beams for the analysis of carbon nanotubes", *J. Appl. Phys*, vol. 103, 023511, 2008

[25] A.A. Ahmadi Asoor, P. Valipour, S.E. Ghasemi, D.D. Ganji, "Mathematical modelling of carbon nanotubes with fluid flow using Keller box method: a vibrational study", *Int. J. Appl. Comput. Math.*, DOI: 10.1007/s40819-016-0206-3, 2016

[26] A.C. Eringen, "On differential equations of nonlocal elasticity and solutions of screw dislocation and surface waves", *J. Appl. Phys.*, vol. 54, pp. 4703-4710, 1983

[27] N. Herisanu, V. Marinca, "An optimal approach to study the nonlinear behaviour of a rotating electrical machine", *J. Appl. Math.*, ID 465023, 2012

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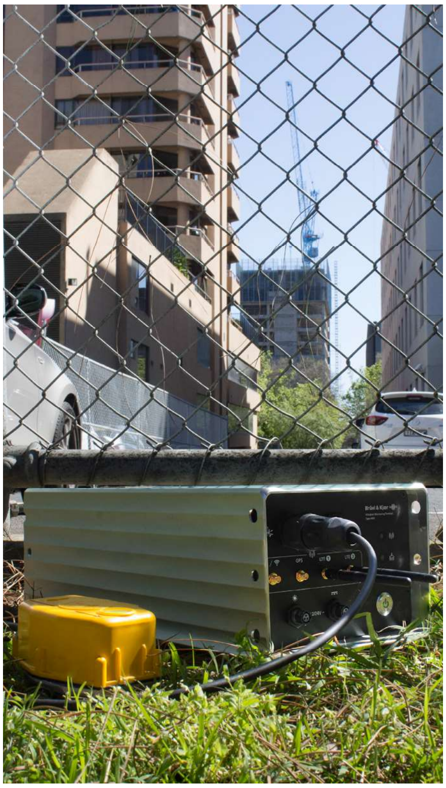

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MODELING EFFECTS OF LIVE HUMAN EXCITATION TO THE MECHANICAL SYSTEM USING THE DISCRETE-TIME FOURIER SERIES

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Abstract - *Accidental actions which cause oscillations of mechanical systems in supporting structures are a class of dynamic tasks of specific scientific and professional work of the current safety of people and equipment. This paper shows the determination of mathematical models of physical excitation force that can create a man with his power, based on the discrete-time Fourier transformation. For experimental verification of the model was made the test platform for the measurement of force, which hopping people created natural oscillatory impulse. The paper shows the results of individual and group experimental testing live human forces as the cause of accidents and dangerous effects on the structure. The end of the work shows the application of the excitation of the human operation of a malicious nature (the high transport machine - crane) which shows the application Fourier model to simulate one incident. For this purpose, the transient FEM analysis and eigenvalues predetermined modal analysis are used. The paper is illustrated with photographs of experimental tests of the dangerous human impulse in several working machines (objects).*

1. INTRODUCTION

A long time is already known that living human force can cause critical states of facilities. The pulse effect in the form of rocking, swinging or synchronous bouncing (by dynamic excitation force) may increase mechanical action on the structure. That action can be enough to cause an incident (breakdown) in the form of overturning in the case of systems with limited stability. Especially, living human effect can be risky when it takes place in a resonant mode with more frequently synchronized people.

National experimental investigations carried out on elastic support structures of cranes in the last 30 years [1-4] show the dynamic sensitivity in various forms of inattention when handling. Inattention is often manifested through risk, incidents and damages. Risk caused by small mechanical work, i.e. by energy that people can create only from their physical strength is particularly interesting. In 1996 Great Britain adopted the standard BS 6399-1 BS [5] which relate to live actions. Littler [6] introduced in 2003 a mathematical model of live action induced by jumps and impacts. In 2006 Sahnaci and Kasperski presented a model of random loads induced by walking [7]. Behnia et al. [8] provide a review of several load models: Half-Sine Pulse (proposed by Bachmann and Ammann 1987), Dynamic loading using Fourier series (based on summing up a group of sine waves of various frequencies, magnitudes and phases, convenient for human loading) and expansion to crowds, according to the relation:

$$F_v(t)N = C(N)\cdot Q\cdot\left[1+\sum_{n=1}^{k}\alpha_{n,v}\cdot\mathbf{sin}\left(2\pi\cdot f\cdot n\cdot t+\Phi_{nv}\right)\right] \quad (1)$$

The model of crowd loads is defined by the magnitude of applied load at time for a crowd of N people $F_v(t)$ as a product of the coordinating factor C(N) of a size N group (the event is based on the crowd load of at least 20 persons), Q is the weight of associated crowd, k is the DLF number, α_n is the n-th rhytmic (DLF) coefficient which is represented by different people in different activities, f is the activity frequency and Φ_{nv} is the phase lag of the n-th harmonic of the load. The ISO 10137 standard (2007) defines the rhytmic coefficient as α_n=0.8 maximally and α_n=0.5 minimally in the first harmonic. The values of this coefficient are α_n=0.67 according to BS 6399-1 (1996) and α_n=0.75 according to Bachmann and Ammann (1987). There are also newer load models based on random actions (Vakulenko and Cherkai [9], 2010) and the duration of the contact between feet and the base (Silva and Thambiratnam [10], 2009).

This paper, in the further course, first exposes the original experimental research on a simple mechanical model (stand) and then on a real supporting structure of the bridge crane with trail. The results of numerical simulation of the bridge crane and comparison with the results of the experiment are given at the end of the paper.

2. A SIMPLE EXPERIMENT WITH LIVING HUMAN FORCE

By starting research effect of living human force on facilities one question was asked: How does the disturbance force created by one person's energy look like? For this purpose a test platform to measure the vertical bouncing force by people (several persons) was made, Fig. 1. The test platform is equipped with a force transducer HBM U2A and acceleration sensor Philips. The measurement was performed with full six-channel measuring bridges with feedback between the amplifier and sensors. Measured results (Measurement no.15, TEST1_014, April 22, 2015, start time 11:34 AM) of the bouncing force by an adult male person (Measurement no.3) without separation from the upper surface of the platform and medium intensity of effect between limits Rp=1÷3 are shown on the diagrams in Fig. 2.

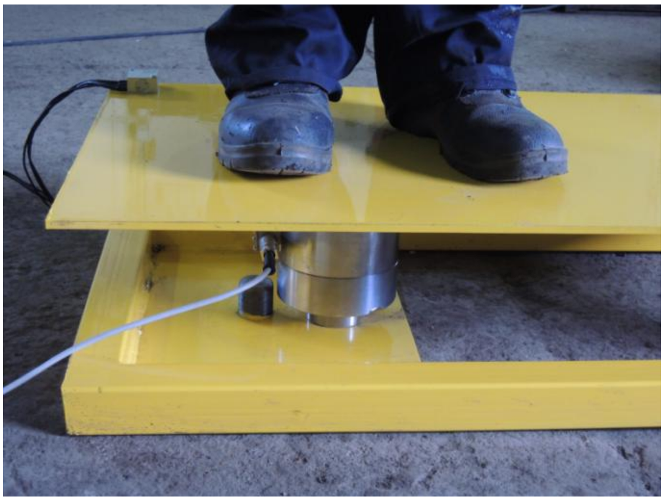

Fig. 1 *Experiment: Test stand with a person and two sensors*

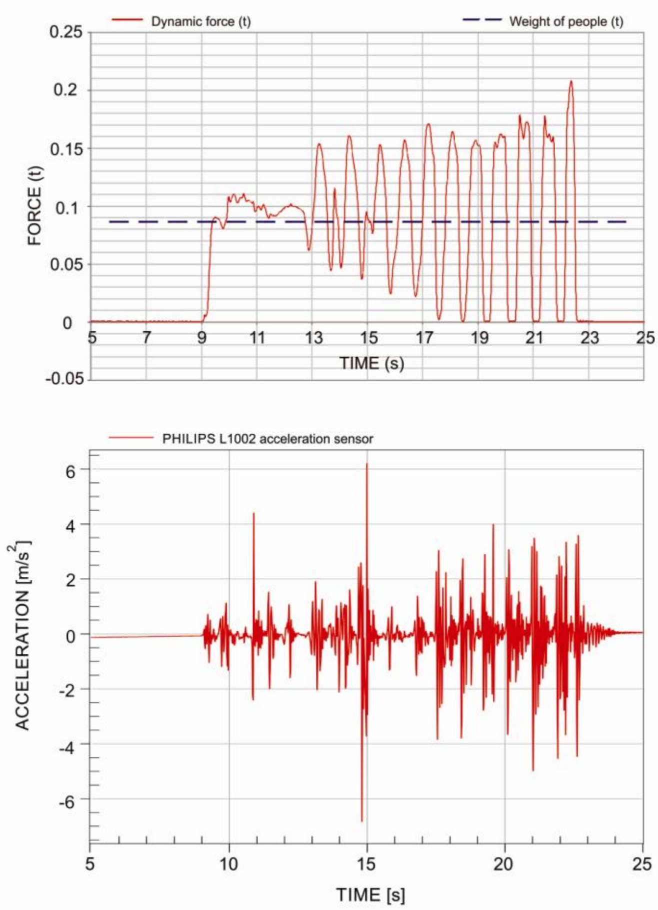

Fig. 2 *Experiment No.15: Living human force F(t) [ton] and test stand acceleration a_y [m/s^2] in the vertical direction caused by vibration*

On the force diagram (Fig. 2) one can see that the dynamic change of force has a periodic harmonic character and intensity conditioned by the intention to obtain uniform scope of excitation between 9th and 23rd second of the experiment. This resulted in the maximum ratio of dynamic and static force (dynamic coefficient) of 2.4 and maximum acceleration of the tread surface of 6.85 m/s^2. This and other experimental results [11] with more persons on the platform indicate the possibility for theoretic modeling of live human force using Fourier series as proposed by Bachmann and Ammann 1987. This model is based on the summation of individual sine semi-waves or full waves caused by various excitation frequencies. One can see that the living human force has not a constant amplitude but that it is conditioned by the individual development (case). Consequently it is necessary to introduce an additional function amplitude of the resulting force which is caused by the process of growth force - excitation of the facility from hibernation.

3. SETTING THE THEORETICAL MODEL OF THE LIVING HUMAN FORCE

In order to implement previous observations about the live human force character on a real object – elastic supporting structure of crane, the following assumptions have been made:

- Modelling the incident as a dynamic event caused by a malicious action on heavy lifting machines.
- Modelling a malicious dynamic excitation on the main crane carrier according to the performed experiment, Figs. 2 and 3.
- The excitation is created by the effect of two persons (i=2) as their own live force at bouncing on the main carrier of the crane.
- The excitation is characterized by the synchronicity of individual living actions $F_i(t)$.
- Oscillation mass M_i of people who caused the excitation were known and frequency Ω was measured.
- The damping force is proportional to the nearest modal frequency (ω_{17}) – the excitation frequency (Ω).
- The excitation is characterized by a monotonous growth and reducing amplitudes of stress according to each experiment performed, Fig. 3.
- The duration of excitation in the experiment and number of oscillations to the rest of process is known.
- The amplitude of vertical translation due to the excitation of forced living force is experimentally determined.
- Boundary conditions of the experiment are known: Initial excitation time t_0, end time of excitation t_K=t_7, and characteristic excitation times: t_1, t_2, t_3, t_4, t_5, t_6, t_7.

According to Eq.(2) on the basis of previous nine features of the crane bouncing process and dynamic load described by Fourier series [8], a multiplicative oscillatory linearized function of excitation is set. This multiplicative function $F_M(t)$ is a product of people weight G_0, linearized countur amplitude function $F_L(t)$ and harmonic function $F_H(t)$. Value of the maximum amplitude of excitation human weight force G_0 is determined on the basis of experimental testing and it has individual character. G_0 is the force of a group of people N with individual mases M_N which cause the live human force. In this numerical model with the load of two people, value of the force is G_0=2×1000 N. Synchronization of load is complete i.e. α=1.

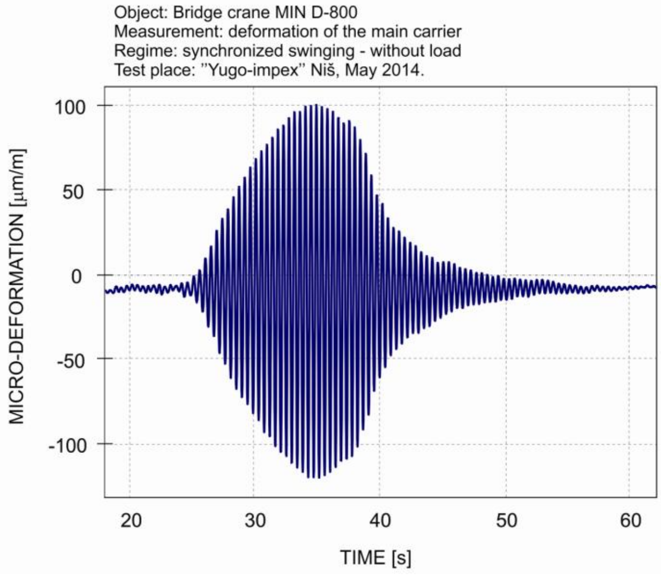

Fig. 3 *Diagram of longitudinal strain deformation of the main box girder of bridge crane during malicious crane swinging with two persons (Measuring no.11, May 2014)*

$$F_M(t) = G_0 \cdot \left[1 + F_L(t) \cdot F_H(t)\right] \tag{2}$$

In general, the countur amplitude function $F_L(t)$, Fig. 4, is determined by linear approximation of the stress/strain growth experimentally obtained and shown in Fig. 3. This function is formed from seven chronological successive functions $F_n(t)$ (n=1÷7) introduced by Eq.(3). The harmonic excitation function $F_H(t)$ at the forced frequency Ω is represented in further text as the sine function, Eq.(4). The multiplicative oscillatory linearized function of excitation, Eq.(2), then takes a characteristic theoretical form which is shown in Fig. 5.

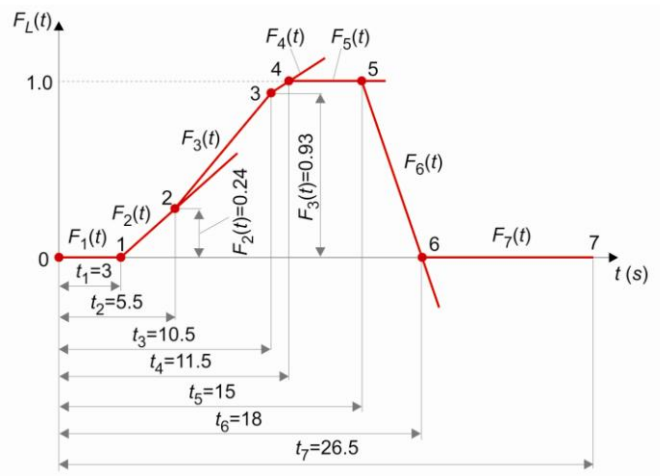

Fig. 4 *The general model of contour amplitude function*

$$F_L(t) = \begin{cases} F_1(t)=0, \quad t=0 \div t_1 \\ F_2(t) - 0 = \dfrac{F_2 - F_1}{t_2 - t_1}(t - t_1), \ t=t_1 \div t_2 \\ F_3(t) - F_{2(t=t2)} = \dfrac{F_3 - F_2}{t_3 - t_2}(t - t_2), \ t=t_2 \div t_3 \\ F_4(t) - F_{3(t=t3)} = \dfrac{1 - F_3}{t_4 - t_3}(t - t_3), \ t=t_3 \div t_4 \\ F_5(t)=1, \quad t=t_4 \div t_5 \\ F_6(t) - 1 = \dfrac{F_6 - 1}{t_6 - t_5}(t - t_5), \ t=t_5 \div t_6 \\ F_7(t)=0, \quad t=t_6 \div t_7 \end{cases} \tag{3}$$

$$F_H(t) = \mathbf{sin}(\Omega \cdot t) \tag{4}$$

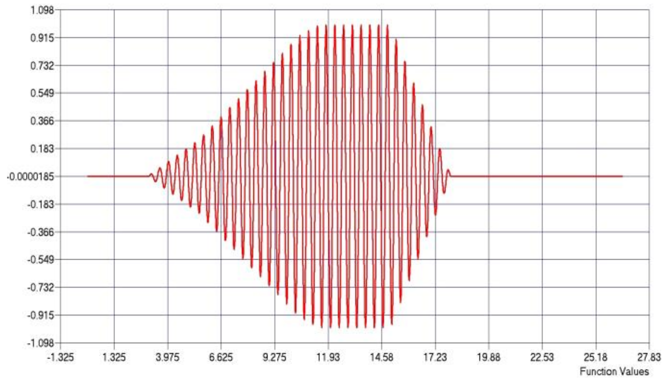

Fig. 5 *The multiplicative oscillatory linearized function of excitation. The parameters of applied function $F_L(t)$: t_0=0 s, t_1=3 s, t_2=5.5 s, t_3=10.5 s, t_4=11.5 s, t_5=15 s, t_6=18 s, t_7=26.5 s, F_2=0.24, F_3=0.93, F_6=0, Ω=2.306 Hz*

4. NUMERICAL SIMULATION

Transient analysis has nonlinear character. So, according to the author's choice, an acceptable solution may be required using the Newmark method of time integration. In this way, the system of equations for the geometric nonlinear structural dynamics can be written at time t+Δt as:

$$\mathbf{M}\,{}^{t+\Delta t}\ddot{\mathbf{q}}^{(k)} + {}^{t+\Delta t}\mathbf{C}\,{}^{t+\Delta t}\dot{\mathbf{q}}^{(k)} + {}^{t+\Delta t}\mathbf{K_T}\,{}^{t+\Delta t}\mathbf{\Delta q}^{(k)} = \\ = {}^{t+\Delta t}\mathbf{f_{ext}} - {}^{t+\Delta t}\mathbf{f_{int}}^{(k-1)} \tag{5}$$

Where: **M** is the matrix of inercial coefficients of the model, **C** is damping matrix, $\mathbf{K_T}$ is the tangential matrix of stiffness, Δt is the increment of time, k is the ordinal number of numerical iterations.

4.1 Bridge crane modelling

The discrete FE model of bridge crane is based on a real crane of manufacture type MIN-D800 (the owner: company JUGOIMPEX-Niš) with the range of L=30 m, carrying capacity of Q=5 t and mean structural elasticity on the bending. The crane is characterized by the total mass of 17.2 t and carriers of medium stiffness (L/f=424>250, where f is the deflection of the mid-range). For the needs of this research the authors developed a model including the crane trail and support masts. The numerical model has 15786 finite elements and 87982 algebraic equations (degrees of freedom). The model was solved using MSC NASTRAN software, Fig. 6. Very high fidelity of the model in relation to the physical object (crane) is enabled by introducing the crane trails and support masts.

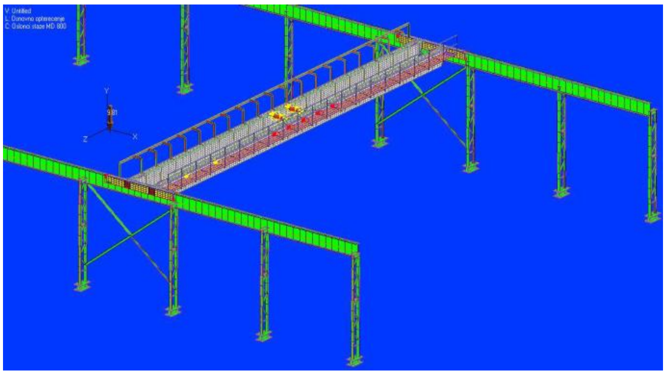

Fig. 6 *FEM model of the bridge crane MIN-D800 with trail and supports in the numerical simulation*

4.2 The solution of simulation for living force excitation

After numerical integration of differential equations (in the direct transient analysis) which describe the numerical model of the bridge crane, a response of the main crane carrier on the living force excitation is obtained on the basis of the adopted step integration of 20 ms in a separate time interval from Fig. 5. The solution of numerical integration in the form of vertical translation of the node in the middle of the carrier range in time function is shown in Fig. 7. The excitation was modeled as an action of two people on the carrier by synchronous bouncing at its lowest eigenfrequency.

The first group vibration from the diagram in Fig. 7 (up to 10 seconds) is caused by calming the carrier due to a static force introduced in the model – the own weight of people (exciter).

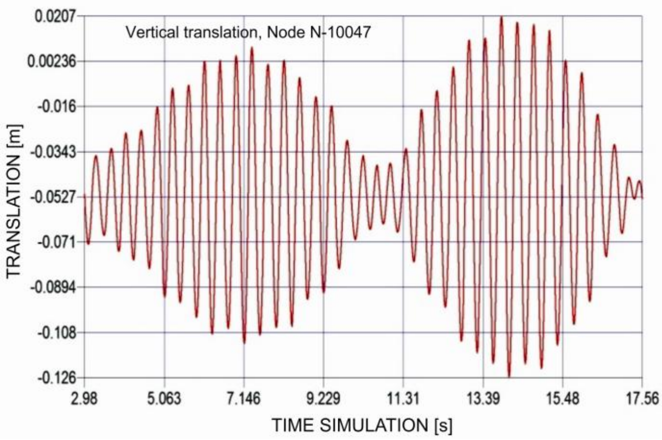

Fig. 7 *Bridge crane D800: Numerical vibration of the main carrier in the node on the middle of the range*

5. EXPERIMENTAL VERIFICATION

For the observed case of incident, an experiment on a real object was carried out. The vertical vibration of the middle of the carrier as an experimental result is shown in Fig. 8. The vibration shape numerically computed, Fig. 7, is very similar to the experimental translations of the mechanical system with damping, Fig. 8. That small difference lies in a longer time interval, that is, a larger number of vibrations. The numerical computed amplitudes of vibration in total amount $2A_N$=0.14 m (Fig. 7) while the experimentally measured amplitudes amount $2A_E$=0.118 m, (Fig. 8). The difference is caused by the influence of the damping due to which the experimental translation has less amplitude.

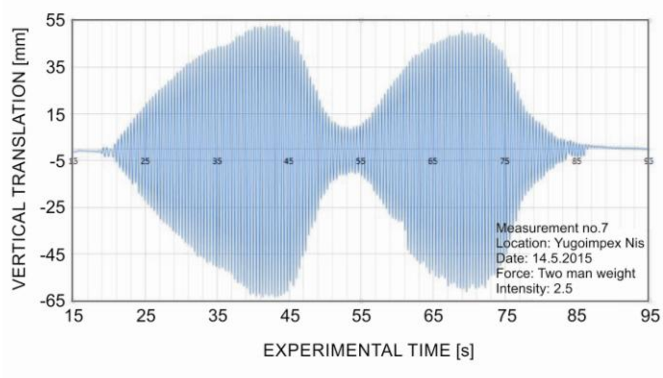

Fig. 8 *Experiment (MIN D800): Simulation of the main carrier vibration in the node on the middle of the range*

Verification of the simulation model of the crane MIN-D800 is performed on the basis of the comparison of two static sizes, **stress** and **deflection** due to the lifting a weight of 4 tons as well as a dynamic size - **vibration frequency**, [12]. Thus, a high approximation of the results is obtained. The longitudinal static stress numerically computed at the load of 4 tons is 1.945 kN/cm^2 while the experimentally measured mean stress is 2 kN/cm^2. The total deflection numerically obtained is 0.0704 m (model D800/ver.7) while the deflection of load is 0.0201 m. The deflection of 0.019 m is measured experimentally at the load of 4 tons (model D800 /ver.4).

In the modal (numerical) analysis that was performed in the second step the eigenfrequencies and eigenvectors of vibration are obtained, [12]. The eigenfrequency of free bridge crane vibration without damping is numerically obtained and amounts ω_{17}=2.306 Hz (without load). This eigenfrequency numerically computed is then experimentally verified using the frequency determined from the diagram in Fig. 8 on the basis of the period of vibration T=0.4602 s (eigenfrequency ω_{17}=2.173 Hz, Table 1).

Table 1 *Comparison of deflection and eigenfrequency*

Type of analysis	Load deflection Δy_Q [m]	Eigenfrequency ω [Hz]
Nonlinear static	0.0201	×
Normal modal	×	2.306
Experimental	0.019	2.173
Relative deviation	5.79%	6.12%

The crane model is experimentally verified using the three checks (stress, deflection, eigenfrequency) which further enabled its application in the modelling dynamic simulation. The model has agreed with the experiment only after introducing the crane trail supported by the vertical truss carriers in the simulation. Fig. 9 shows a characteristic mode shape of the simulation model at a frequency which can easily create a couple of people.

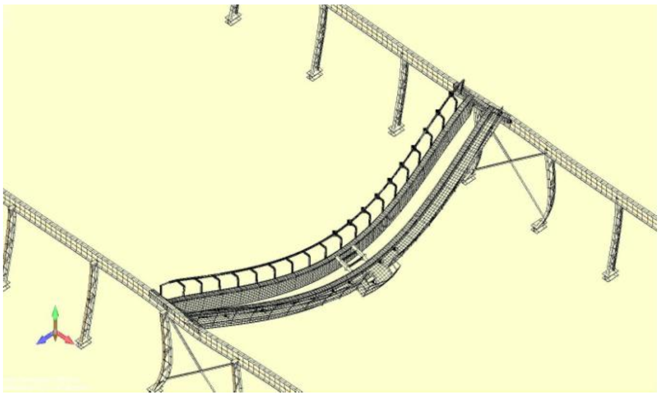

Fig. 9 *Modal analysis: Mode-17 (ω_{17}=2.306 Hz) as a vibration form of crane MIN-D800*

6. CONCLUSION

1) These studies developed and verified models that define practical simulation procedure of incidents on the cranes. The proposed simulation models are rational regarding the number of degrees of freedom and the practical realization using a standard computer on today's level. It allows all shapes and durations of simulation as well as various modalities of external incidental influences. The proposed models can be applied in the practice for individual technical solutions.
2) For bridge cranes with longer ranges and small carrying capacity (30 m/5 t) as well as small relative static stiffness of the main carrier (l/y<400) has been observed that there is a sensitivity of the structure on the occurrence of higher amplitudes of vibration after the experimental investigations and simulations. The risk lies in a relative small energy of excitation which only a few people have. The incident can cause the fall of the winch off the track.
3) The developed numerical models give the results of dynamical simulations for a group of separated dynamical influences. The same models can be used for further investigations of other new types of incidental effects using primarily numeric as a basis for the effective realization.
4) Malicious effects of people can cause significant vibrations, increase of stress and deflection of structure and even, in the case of bridge crane, the winch fall off the track. It is necessary to increase lowest stiffness of the crane carriers and introduce monitoring of the exploitation, considering that a relative small human force can create a very risk state of the stability of elastic objects. In order to protect the human potential of unforeseeable influences it is necessary to apply more frequent remote control of cranes and remove man further from the crane.

REFERENCES

[1] M. Jovanović, "Supporting structure level luffing system and driving mechanisms resistance of portal-jib cranes optimization", Doctor's work, Mechanical Faculty University of Niš, 1990.

[2] M. Jovanović, D. Mijajlović, Lj. Petković, J. Jovanović, "Deseleration time of vibration-parameter of geometric supporting structures synthesis", Faculty of Occupational Safety Niš, Int. Conference Noise and Vibration, 2000.

[3] M. Jovanović, G. Radoičić, "Dynamical response of structures to malicious and random actions", The 5th Int. Conf. "Transport and Logistics – TIL 2014", Niš, 5th-6th June 2014, pp. 51-54.

[4] M. Jovanović, G. Radoičić, P. Milić, "Dynamic sensitivity research of portal-rotating cranes", XIX Int. Conf. on "Material Handling, Constructions and Logistics – MHCL'09", Belgrade, 15th-16th October 2009, pp. 61-66.

[5] British Standards Institution, BS 6399-1:1996, Loading for buildings, British Standard Inst., 1996.

[6] J.D. Littler, "Frequencies of synchronized human loading from jumping and stamping", The Structural Engineer 22, pp. 27-35, 2003.

[7] S. Sahnaci, M. Kasperski, "Random loads induced by walking", Structural Dynamics, EURODIN-2005, Vol. 1, pp. 441-446, Millpress, Rotterdam 2005.

[8] A. Behnia, A.K. Beng Hong, M.M. Shabazi, N. Ranjbar, N. Behnia, M.R. Vafaei, "Finite element analysis of high modal dynamic responses of a composite floor subjected to human motion under passive live load", LA Journal of Solids and Structures, Vol. 10, No. 3, pp. 601-630, 2013.

[9] S.A. Vakulenko, M.V. Cherkai, "Destruction of dissipative structures under random actions", Theoretical and Mathematical Physics, Vol. 165, Issue 1, pp. 1387-1399, 2010.

[10] S.S. De Silva, D.P. Thambiratnam, "Dynamic characteristics of steel–deck composite floors under human-induced loads", Computers & Structures, Vol. 87, Issues 17–18, pp. 1067–1076, 2009.

[11] M. Jovanović, G. Radoičić, "Experimental research of dynamical live-load on the platform in company Mediana-Niš", Internal Report of the Project TR35049, Faculty of Mechnical Engineering Niš, 2015.

[12] M. Jovanović, G. Radoičić, V. Jovanović, V. Tomić, "Synchronic excitation – a type of malicious dynamic action", XXI Int. Conf. on "Material Handling, Constructions and Logistics – MHCL'15", Vienna, 23th-35th September 2015, pp. 207-210.

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EIGENFREQUENCIES OF WATER TOWERS

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Abstract - *The paper, due to the potential occurrence of resonance under the action of periodical impulses, for instance - due to earthquakes, behavior of water towers is analyzed. Eigenfrequencies of water towers depend both on the structure of the water tower in question and the water in it and on the soil-foundation interaction. Namely, eigenfrequency of the tower in terms of vibrations caused by the impulse dependent on the elastic properties of soil beneath the foundation footings and eleastic characteristics of the structure supporting the tank. For the numerical example of aeroelastic frequency research for the first four oscillation tones. Also, for various levels of water in the water tower cone, logarithmic decrement vibration damping values were drawn depending on the level of water in the water tower.*

Key words – *water tower, eigenfrequencies, soil characteristics, logarithmic dampening decrement.*

1. INTRODUCTION

If a structure made of elastic material, or one founded in the elastic soil, such as a water tower, is momentarily displaced from its normal position, by a shock or by a sudden application or removal of a force, the elastic forces in the soil on which it rests, and in its elements are no longer in equilibrium with the external forces, which leads to occurrence of vibrations. Disturbances of elastic equilibrium can be caused by earthquakes, explosions, operation of machinery, street traffic, driving of piles, etc. The magnitude of the equilibrium disturbance, caused by a single impulse, can be expressed either by the intensity of the force causing the disturbance or the distance by which the force displaces the structural center of gravity from its equilibrium. The force is called the perturbation force, and the displacement it creates is called the initial displacement.

If the elastic support of a rigid system is such that the system can vibrate only parallel to given axis or in a plane around an immovable axis, the system is considered to have two or more degrees of freedom. The degree of freedom is equal to the number of coordinates required to determine displacement of a body. In the most general case, the motion of a solid body can be decomposed to three translatory and three rotational components of motion. Each of these components can be specified by a number. Accordingly, a solid body can have six degrees of freedom, at the most. Since the mutual position of elements of such a system remains invariable, the system is called the simple system (single mass system). If, however, the system consists of several solid bodies which are mutually connected with relatively flexible elements, it will be a complex (multiple mass) system. The vibrations caused by a single impulse are called free vibrations. An interval between two successive passages of any particle thorough the most distant position in a given direction is a natural or eigenperiod of vibration for that direction. Elastically supported rigid system with one degree of freedom has only one eigenperiod. If the system has several degrees of freedom, eigenfrequencies of components of its free vibrations can be different.

Contrary to the free vibrations, which are caused by one shock, vibrations caused by a random periodical impulse are called the forced vibrations.

2. EIGENFREUENCIES OF A WATER TOWER

Due to the potential occurrence of resonance, effects of periodic impulses, for instance, impulses produced by heavy street traffic or earthquakes, on the structures and their foundations depend to a great extend from the ratio of impulse frequency and free frequency of buildings. In order to illustrate the effect of elastic properties of foundation soil on the eigenfrequency of the buildings resting on it, we will examine vibrations of a water tower presented in Fig 1, under the effects of a single impulse. The tower rests on four foundation footings, and the impulse acts on the tank, in the horizontal direction, in one of the symmetry planes of the water tower. The most of the mass is concentrated in the tank. The potential of relative displacement of the center of mass of water in respect of the center of mass of the tank itself, severely complicates the problem (Ruge, 1938). For the purpose of simplification according to Wiliams (1937), the tower can be observed as a simple system (single mass system) if water is attributed only around three quarters of its true weight. Therefore, the mobility of water in the tank is ignore. Similarly, the weight of the tower structure which transfers the weight of the tank onto the soil is ignore, and it is assumed that the effective weight of W tank (the tank itself plus tri quarters of weight of water) is concentrated in the center of gravity O_g of the tank.

The eigenfrequency of the tower, in terms of vibrations produced by the impulse depends on the elasticity of the soil beneath the foundation footings and elastic properties of the structure supporting the tank. It is possible, in extreme case, that the displacement due to the elastic soil deformation are negligibly small in comparison with displacements due to the elastic deformations of the tower structure. In that case, the tower behaves as a punctual mass located on the upper end of a flexible vertical member, whose lower end is fixed. Such vibrations were studied by Wiliams (Wiliams, 1937). The other extreme possibility is the tower as a rigid structure on the elastic soil. Since we are exclusively interested in the interaction of the foundation soil and the structure, we will consider only the latter possibility.

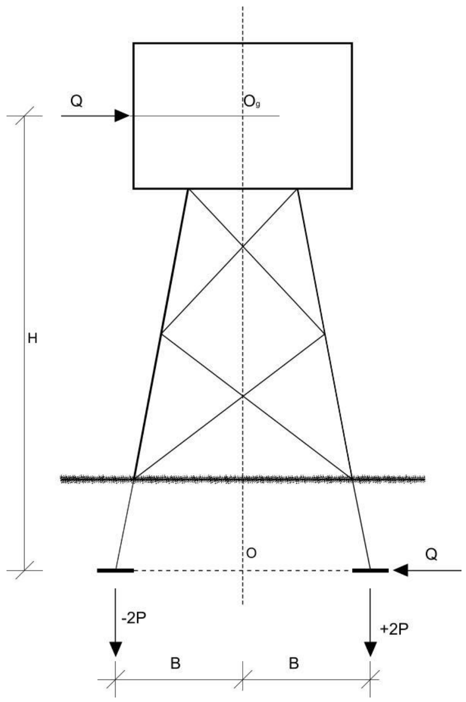

Fig. 1 *Vertical cross-section through the rigid water tower on the perfectly elastic foundations*

If it is assumed that the center O of the tower basis retains its position (i.e. does not move), the tower represents a single mass system, with one degree of freedom. The system can vibrate only parallel to the plane of the figure, oscillating about the horizontal axis which passes through the point O (Fig 1). In addition, we will assume that the coefficient of the dynamic reaction of the foundations soil d_s is determined using the vibrator test. The impulse creates a torque M about the rotation axis through the point O. This moment increases the total pressure on two foundation footings on the right hand side from O for $2P$ for the same force. For an equilibrium, it is required to be:

$$M = 4PB \tag{1}$$

$$P = \frac{M}{4B}. \tag{2}$$

If A is the surface of area of one foundation footing, the pressure P produces vertical displacement of footings which is:

$$\pm \rho = \pm \frac{P}{A}\frac{1}{d_s} = \frac{M}{4AB}\frac{1}{d_s}. \tag{3}$$

The rotation of the tower corresponds to it for the angle

$$\varphi = \frac{\rho}{B} = \frac{M}{4AB^2}\frac{1}{d_s}, \tag{4}$$

About the rotation axis passing through O. Since the tower is rigid, and the axis is immobile, the vertical axis OO_g of the tower will deflect for this angle, and the center of gravity O_g moves for the length:

$$x = \varphi H = \frac{MH}{4AB^2}\frac{1}{d_s}. \tag{5}$$

Since the angle ϕ is small, displacement x is almost horizontal and rectilinear. As the first approximation, we can adopt the assumption that every particle of the tank moves for the same horizontal distance, ϕH. This assumption reduces the problem to the case of linear vibrations. In order to determine the appropriate constant of the spring c_s, we will write:

$$M = QH \tag{6}$$

or

$$Q = \frac{M}{H}. \tag{7}$$

Relation

$$\frac{Q}{x} = c_s\left(gmcm^{-1}\right) = Ak_s, \tag{8}$$

is called the block spring constant, in order to move in the direction of force for the unit. Q is the horizontal force which passes through the center of gravity O_g of the tank. The moment created by the force Q about the rotation axis through the point O is equal to the moment of impulse M. Since we assumed that all the particles of the reservoir simultaneously move for the same horizontal displacement x, the spring constant c_s equals the force which is required to move the tank in horizontal direction for the unit of length (equation (8)). Therefore there is:

$$c_s = \frac{Q}{x} = 4A\frac{B^2}{H^2}d_s. \tag{9}$$

If this value is included into the equation for circular eigenfrequency, the following equation is obtained:

$$\omega_0 = \frac{2B}{H}\sqrt{\frac{A}{W}gd_s}. \tag{10}$$

Eigenfrequency (equation (11)) is the number of cycles in a unit of time:

$$f_0 = \frac{\omega_0}{2\pi} = \frac{B}{\pi H}\sqrt{\frac{A}{W}gd_s}. \tag{11}$$

Static pressure per unit of surface are of the foundation footings is

$$q = \frac{W}{4A}, \tag{12}$$

Therefore:

$$f_0 = \frac{\sqrt{g}}{2\pi}\frac{B}{H}\sqrt{\frac{d_s}{q}}. \tag{13}$$

On the basis of this equation, the following conclusions regarding eigenfrequency of the rigid tower which represents a single mass system, that is, system with one degree of freedom can be drawn. The softer the soil supporting the tower, the lower the frequency f_0. In order to increase the frequency of the tower of the given height, the width of its layout must be increased, or the pressure per unit of surface area of the foundation footings must be decreased.

If the frequency of the periodical impulse of the tower founded on sand is within limits of critical interval for this sand, the impulse will probably increase permanent settlement of the tower, regardless of eigenfrequency of the tower. If it is close to the value of eigenfrequency of the tower, a resonance will occur, which probably causes additional stresses in the structural elements. [1].

It is a general practice, to predict the action of periodical impulses of the structure, such as, for instance earthquake shocks, without calculating the effects of eigenfrequency of the structure on the vibration amplitudes. Of course, it is justified in case if the impulse frequency is significantly higher or lower than the eigenfrequency of the structure. Ignoring this reserve can lead to grave errors.

3. NUMERICAL EXAMPLE OF AEROELASTIC TESTING OF WATER TOWER

Towers are tall structures where horizontal loads are not supported by stays, but flexure of the fixed basis of the tower. The fluid in the tower can have serious dynamic impacts.

Towers are designed by determining the loads in each horizontal direction. The coefficients for tower flexure stresses are 1.5 to 2.5. The considerable loads come from the action of earthquakes.

There can occur resonant zones between the top of the level of fluid in water tower and the tower itself. However, the adequate measures not only separate the frequencies but also increase the dampening. Fig. 3(a) presents the increase of dampening when the level of water is increased in the conical tank (Fig. 2), indicating the potential of doubling the logarithmic decrement. Fig. 3(b) demonstrates than only the fourth tone of the open area has the frequency close to the tone of vibration caused by flexure of the tower itself, and the latter frequency varies for 6% when the tank is full.

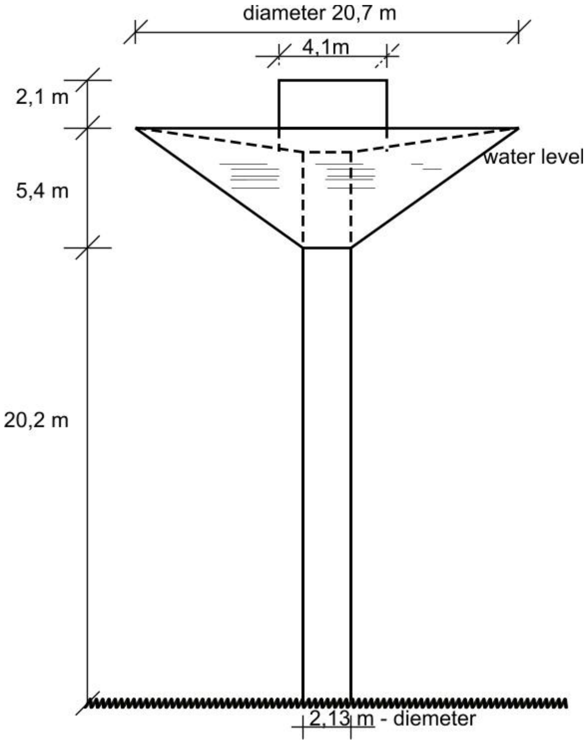

Fig. 2 *Aeroelastic testing of the water tower model. General disposition of the water tower (dimensions of the structure in full scale)* [2]

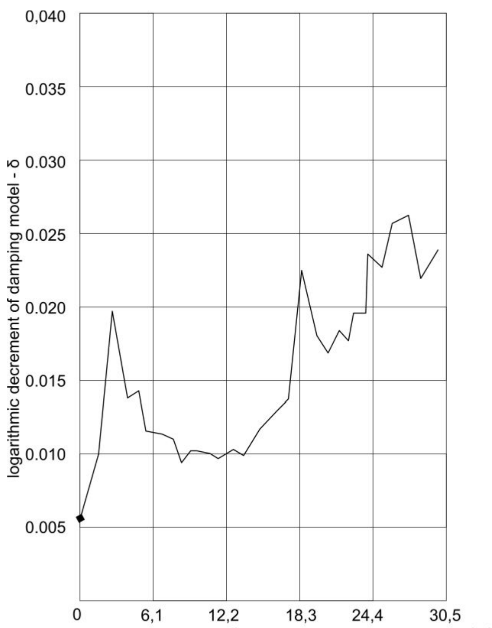

Fig 3 a)

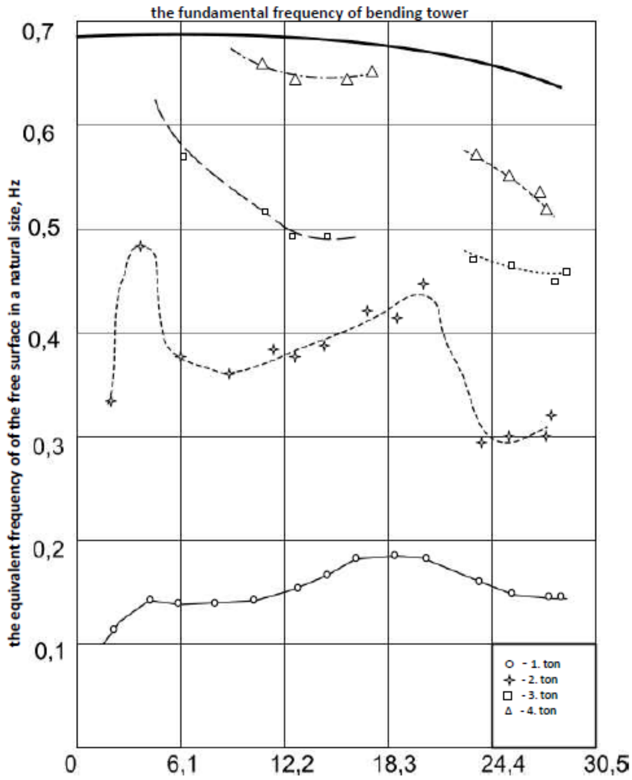

Fig 3 *Aeroelastic testing of the water tower model: a) dampening of water tower; b) frequencies of the open area of the cone*

4. CONCLUSION

Towers are tall structures where horizontal loads caused by vibration of the fluid are supported by flexure of the fixed basis of the tower in the foundations.

There can occur resonant zones between the top of the level of fluid in water tower and the tower itself. Perturbations of elastic equilibrium can be caused by earthquakes, explosions and many other causes. Water tower can be observed as a system with one degree of freedom because the largest portion of mass is the water tower. Because of the possible onset of resonance, the action of the periodical impulses caused by various effects on structures and their foundations depends largely on the ratio of frequency, impulse and eigenfrequency of water tower. Eigenfrequency of the tower, in terms of vibrations produced by the impulse depends on the elastic properties of the soil beneath the foundation footings and elastic properties of the structure supporting the tank. The coefficient of the dynamic reaction of the foundation soil d_s is determined using the vibrator test. On the basis of the derived formulae, eigenfrequency, static pressure per unit of foundation footing are and other necessary parameters are calculated. The softer the soil supporting the tower, the lower the frequency f_0. If the eigenfrequency is not close to the resonance, then it is a common practice that the action of periodical impulses on water tower, such as the earthquake shocks are predicted before calculating the eigenfrequency of the structure to vibration amplitudes.

For the presented conical model of water tower, aeroelastic testing was performed in full scale.

In Fig. 3(a) on the abscise is presented the equivalent depth of water in the cone, and on the ordinate the logarithmic decrement of dampening. In Fig. 3 (b) of the diagram on the abscise, depending on the depth of water in the cone, on the ordinate is presented the equivalent frequency of free water surface in the water tower (in Hertz).

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REFERENCES

[1] K. Terzagi, *Teorijska mehanika tla*, Naučna knjiga, Beograd, 1972.

[2] P. Sachs, *Uticaj vetra na konstrukcije*, Građevinska knjiga, Beograd, 1986.

[3] Ž. Radosavljević, D. Bajić, *Armirani beton 3. Elementi armiranobetonskih konstrukcija-konusne ljuske*, Građevinska knjiga, Beograd, 1988.

[4] Nacrt pravilnika o tehničkim normativima za projektovanje i proračun inženjerskih objekata u seizmičkim područjima, Savezni zavod za standardizaciju, br. 07-93/96, Beograd, 1987.

[5] S. Zdravković, „Postupak za približno određivanje parametara kritičnog opterećenja za konzolni nosač“, *VIII Kongres SDGKJ*, Cavtat, 1987, str. 235-240.

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DETERMINATION OF THE PRECISE EQUIPMENT FOUNDATION DISPLACMENT CAUSED BY SEISMIC EXCITATION VIBRATIONS

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Abstract – *The paper presents protection of machinery and instruments from the undesirable vibrations of their base from the action of earthquakes. A special attention is paid to the protection of medical-hospital equipment. It is often the case that the high-rise structures withstand the earthquakes well, but the equipment in them gets destroyed because it was not made for the same degree of seismic risk (earthquake intensity) as the structure of the building. The provisions of the Code for construction of high-rise buildings in seismic areas providing for the protection of elements and equipment are mostly ignored, as in case of the medical care buildings. Hospitals, as the most important institutions caring of human health, must be ready to provide help at all times, and thus during and after catastrophic disasters such as earthquakes. In those situations, it is important that the equipment in them is undamaged and serviceable, and that the medical personnel is not injured and ready to commit to helping people.*

Key words – *displacement amplitudes, foundation, precise equipment, vibrations, seismic excitation.*

1. INTRODUCTION

When designing structures, it is necessary to provide their functionality, rationality and safety under all potential loads, and thus under earthquake effects. When determining the earthquake load, there is high uncertainty of data which are most often defined in the statistical sense of the probability of occurrence. In the similar sense, the resistance of a building can be defined in the statistical sense, but an absolute reliability cannot be achieved. Yet, with the increase of resistance of the building to the earthquake action, a low probability of collapse can be achieved, i.e. sufficiently high safety. On the other hand, the increase of safety increases financial cost of the structure, and raises the question what degree of safety is acceptable. It is not only a technical but a socio-economic question, and it has remained unanswered yet. It is generally accepted that total protection from earthquake is not economically justifiable for the most of structures whose functioning is important during and after earthquakes, such as hospitals, power plants, firefighting service buildings, water supply facilities, etc. All the previous statements refer mostly to protection of the building structure itself. However, it is often the case that the constructed hospitals, factory halls and similar structures withstand an earthquake well, but equipment and devices in them get destroyed because they were not made for the same degree of seismic risk (earthquake intensity) as the building structure [3]. The section of the Code which regulates protection of the building structure is mostly adhered to in practice. For that reason, the structures withstand an earthquake well, but equipment and devices in them get destroyed because they were not made for the same degree of seismic risk (earthquake intensity) as the building structure. The provisions of the Code providing for the protection of elements and equipment in or on them are mostly ignored.

2. PRECISE MACHINERY AND MEASURING INSTRUMENTS

In order to have a proper operation of machinery and instruments, protected from the undesirable vibrations of their base, the machinery foundation displacement amplitudes and limit values of permissible frequencies must be within a permissible range, so it is necessary to determine correlation between these parameters. This section will lay out the procedure proposed by Klatz [5], on the example of a tooled machine.

It is assumed that the tooled machine represents a system with one degree of freedom, supported on an elastic, oscillating base, fig.1.

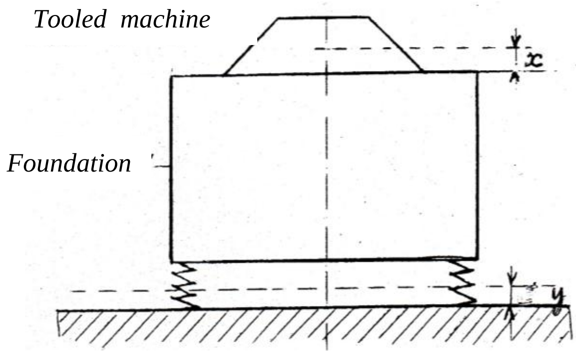

Fig. 1 *Diagram of the tooled machine oscillation due to the motion of the base*

Let the base oscillation be defined with the form equation:

$$y = A\sin\omega_g t \tag{1}$$

And the system oscillation (working machinery with foundations) can be presented by the equation:

$$x = A\frac{1}{1-\left(\frac{\omega_g}{\omega}\right)^2}\sin\omega_g t \tag{2}$$

Designations in these equations are:

y – base displacement,

A – base displacement amplitudes,

ω_g – excitation circular frequency of the base (s^{-1}),

x – oscillating system displacement (working machinery with foundations),

ω – circular eigenfrequency of the oscillating system (s^{-1}),

If it is assumed that the metal working tool is tied to the reduced mass of the tooled machine, and the worked specimen is tied to the base, the mutual displacement of the tool and the specimen will be determined by the expression:

$$\eta = x - y \tag{3}$$

If the equations (1) and (2) are introduced to this equation, the following is obtained:

$$\eta = A\frac{1}{1-\left(\frac{\omega_g}{\omega}\right)^2}\sin\omega_g t - A\sin\omega_g t = = A\sin\omega_g t\left(\frac{1}{1-\left(\frac{\omega_g}{\omega}\right)^2} - 1\right) = = A\frac{1}{\left(\frac{\omega_g}{\omega}\right)^2 - 1}\sin\omega_g t \tag{4}$$

where η expresses relative displacement of the tool and the worked specimen in the random moment *t*. Maximum relative displacement will occur at $\sin(\omega_g t) = 1$, and it amounts

$$\eta_n = A\frac{1}{\left(\frac{\omega_g}{\omega}\right)^2 - 1} \tag{5}$$

If conditioned boundary values are included to this expression, for $\eta_n=\eta_{doz}$ and for $A=A_{doz}$, then the equation (5) can be written in the following form:

$$\eta_{doz} = A_{doz}\frac{1}{\left(\frac{\omega_g}{\omega}\right)^2 - 1} \tag{6}$$

$$A_{doz} = \eta_{doz}\left[\left(\frac{\omega_g}{\omega}\right)^2 - 1\right] \tag{7}$$

Equation (7) establishes correlation between the permissible values of foundation displacement amplitude A_{doz} and permitted difference of displacement η_{doz} between the tool and the worked specimen.

The value of η_{doz} according to Klatz [4] amounts to:

- For small tooled machines η_{doz}= 4 mm
- For medium tooled machines η_{doz}= 2 mm
- For heavy tooled machines η_{doz}= 1 mm.

Parameter ω, which represents lower frequency of eigenfrequencies of the tooled machines, as the most dangerous form for precision of its operation, should be determined experimentally when designing the foundations. The research demonstrated that this parameter ranges between

- $\omega = 200\ s^{-1}$ for the heavies types of tooled machines, for instance for machining gear cogs,
- And ω = 500-600 s^{-1} for the lightest types of machines.

The research further demonstrated that for the operation of precise working machines, the base frequencies exceeding 50-60 s^{-1} are dangerous. The listed data indicate that for the precise tooled machines, the foundations with vibro-insulation should be used (steel helical springs, rubber elements) when there are high-frequency sources of vibrations with circular vibrations higher than the mentioned limit values, such as mechanical hammers, presses, heavy road traffic, etc.

When designing passive vibro-insulation, the relationship of excitation and eigenfrequency $\lambda_1=\omega_g/\omega$ is determined depending on the necessary magnitude of transmission coefficient μ, according to the expression:

$$\lambda_1 = \frac{\omega_g}{\omega} = \sqrt{\frac{1+\mu}{\mu}} \geq 4 \tag{8}$$

Where the transmission coefficient in the vertical direction is defined with the expression:

$$\eta = \frac{a}{A} \tag{9}$$

In this expression:

a – is the amplitude of vertical oscillations of the center of gravity of the system being isolated (machine with the foundations, forced oscillations),

A – is the amplitude of vertical oscillations of the structure that supports it (base).

The amplitude of vertical oscillations of the system which is passively insulated from base vibrations, can be determined from the expressions (8) and (9).

$$a = A\frac{1}{\lambda_2 - 1} \tag{10}$$

The mentioned parameters should be used when designing foundations to be passively isolated from the base vibrations.

Passive isolation of machines, which are exposed to temporary or momentary shocks, must have elements for oscillation damping, with the coefficient of inelastic resistance γ_m=0.04 to 0.05. Such value is achieved using dampener with viscous friction, air dampener, combined vibro-isolators made of steel springs and rubber elements. When making combined vibro-isolators, it is recommended to use porous rubber of the dynamical model of elasticity E= 50 to 150 N/cm^2 and permissible stresses of 2 N/cm^2 [5]. On the diagram in figure 2 are included frequency zones where application of the mentioned spring elements is possible.

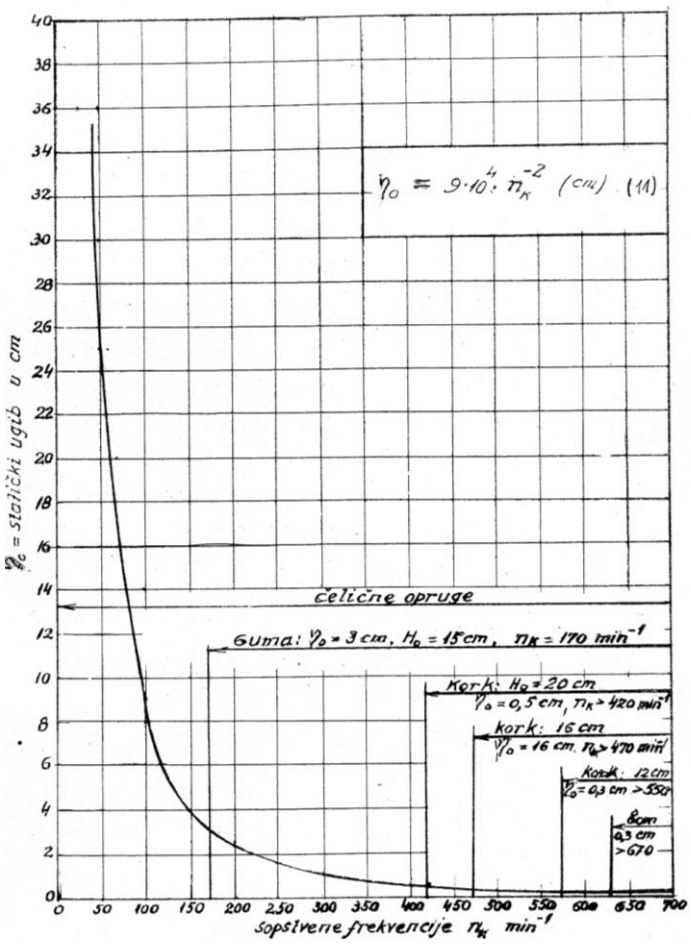

Fig. 2 *Dependence of deformation η_0 at static load from the frequency n_k for various materials (steel, rubber, cork)*

3. EXCERPTS FROM THE CODE FOR PROTECTION OF STRUCTURAL ELEMENTS FROM EARTHQUAKE EFFECTS

According to the article 35 of the Code [1], the impact of seismic forces on the structural elements is calculated according to the expression:

$$S = S_k K_e G_e \qquad (11)$$

where:

K_s =(0.33-1.0) – is the coefficient of seismic intensity according to the article 24 of the code,

K_e =(2.5-10.0) – coefficient according to the article 36 of the Code,

G_e – weight of the structural element for which the seismic force is calculated.

According to the article 37 of the code [1], anchoring of the equipment in the building structures whose displacement or toppling can endanger human lives or create damage is calculated by the term from the article 35 of the Code [1] with K_e = 10.0 for the purpose of securing equipment from displacement or toppling.

According to the article 38 of the Code [1], the calculation of anchoring of precious equipment whose operation is necessary in the buildings is performed using the method of dynamical analysis of the structure – equipment. To prevent the situation in which, after an intensive earthquake, in a relatively undamaged structure, there is a large portion of damaged and unserviceable equipment, it is necessary to secure both the structure and the equipment with the same degree of safety. Some American research and experience in the earthquakes indicate that due to the amplification effects on some parts of equipment the acceleration amounted to more than 1.0 g in the horizontal and vertical direction of action.

3.1 Protection of hospital equipment

Hospitals, as the most important institutions caring of human health, must be ready to provide help at all times, and thus during and after catastrophic disasters such as earthquakes. In those situations, it is important that the equipment in them is undamaged and serviceable, and that the medical personnel is not injured and ready to commit to helping people.

Medical equipment is very expensive, but more importantly, it is necessary, so those are the main reason to take care about its adequate protection. The methods of protection are varied an relatively simple and cheap. The medical equipment consists of complex necessary life support devices, to unimportant but very precious auxiliary equipment. In case the complete equipment is not preventively protected in an adequate way, there can be horrifying scenes after earthquakes: scenes of overturned and broken equipment on the floor, broken glass, spilled liquids, injured people – both the patients and medical personnel. Equipment after earthquake can be appear good, however, because of internal malfunction or potential damage of auxiliary devices can be unserviceable.

In design of protection of equipment, the same principles and methods as for the design of the structure are implemented. However, none of design methods provides guarantees that the equipment will remain serviceable after the earthquake. In order to secure and establish the actual behavior of equipment during earthquake, it should be subjected to dynamic testing on a seismic platform. Depending on the importance of the equipment which needs to remain operational during and after earthquake, there are the following methods for seismic classification, those being: dynamical test, mathematical analysis, previous experience, assessment of the team of experts ad combination of previous methods. After application of the combined method, with using of literature, all the medical and auxiliary equipment can be systematized according to the types of protection [2]:

1) The massive equipment lying on the floor (boilers, cooking stoves, washing machines etc), regardless of how it is fixed to the floor, must have fenders on all sides. (fig.3).

2) The massive equipment located high above the floor on steel posts (heating oil tanks, large fan in the auxiliary room next to the kitchen, etc.) must have struts among the posts in two planes and the housing must be attached to the walls (fig.4).

3) On the shelves there must be plastic "parapets" in order to prevent objects from falling off to the floor, and it is desirable that they are plastic, and not glass (fig.5).

4) Tall furniture placed next to the walls (refrigerators, phone switchboards, equipment for spirometry etc.) must be attached to the walls to prevent toppling (fig.6).

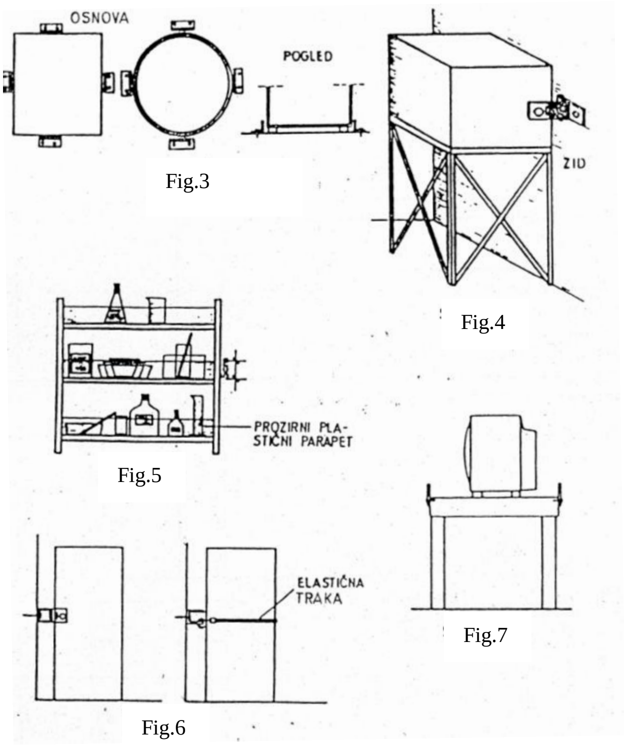

Fig. 3-7

5) On the tables and working surfaces supporting TV sets, monitors, ECG devices etc. must have wooden side boards on the edges. (fig.7).

6) The mobile equipment (monitors, reanimation devices etc..) must have holders in the walls and on the equipment as shown in figure.8.

7) The devices sliding along rail systems (RTG apparatuses, graphoscopes, tomographs etc.) must be checked for fixation of the rails to the floor structure, and the limiters on the ends of rails need to be installed. (fig 9).

8) In case of large ventilation ducts suspended from the ceiling in the kitchen and other rooms, the duct brackets must be placed in two orthogonal planes (fig.10).

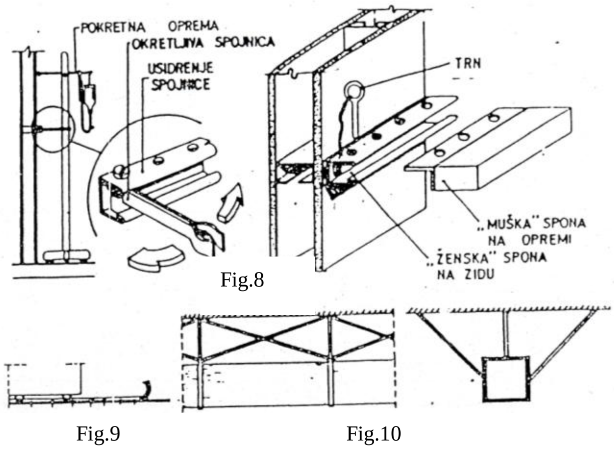

Fig. 8-10

4. CONCLUSION

Regarding that human lives have no price, the equipment in important structures, and especially in hospitals, must be paid more attention then until now. When this concerns equipment necessary for life and equipment which provides limitless and necessary operation work of the system, its reliability must be provided at all costs, both during and after earthquakes. The equipment in future and existing structures must be provided with the identical degree of seismic safety, as it is of special importance for life and health of people.

REFERENCES

[1] "Pravilnik o tehničkim normativima za izgradju objekata visokogradje u seizmičkim područjima", Službeni list SFRJ br.31/1981.

[2] Zbornik radova V Kongresa saveza društva za seizmičko građevinarstvo Jugoslavije, Bled, 1990.

[3] S. Zdravković, "Zaštita objekata od vibracija izazvanih pomeranjem tla za vreme zemljotresa", *XIII YU Savetovanje "Buka i vibracije u životnoj i radnoj sredini".* FZNR, Niš, 1991.

[4] O. A. Savinov, *Savremene konstrukcije temelja mašina i njihov proraču*, Beograd, Građevinska knjiga, 1966.

[5] J. Mandić, *Temeljenje mašina sa primenom elastičnih izolatora oscilacija*, Beograd, Naučna knjiga, 1985.

[6] P. Sachs, *Uticaj vetra na konstrukcije*, Beograd, Građevinska knjiga, 1986.

[7] S. Zdravković, "Aseizmička zaštita u objektima visokogradnje", *XI Naučni skup- međunarodna konferncija "Preventivni inženjering i životna sredina"*, FZNR, Niš, 1995.

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STRATEGY FOR EFFECTIVE MEASUREMENT OF HAND TRANSMITTED VIBRATION AT THE WORKPLACE

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Abstract - *This paper shows stages in which the evaluation of vibration exposure can be broken up: identifycation of a series of discrete operations which make up the subject's normal working pattern; selection of operations to be measured; organization and duration of the measurement, d) estimation of daily vibration exposure time; measurement equipment, location and attaching of accelerometers; measurement of vibration magnitude; sources of uncertainty in vibration measurement; calculation of the daily vibration exposure. In addition, one example of the calculation of daily vibration exposure of workers engaged in the "DalCin" d.o.o. Company is shown.*

1. INTRODUCTION

Operating machinery, such as pneumatic, electric, hydraulic or internal combustion engine-driven chain saw, pecussive tools or grinders, may expose workers to hand transmitted mechanical vibration that can affect the comfort, working efficiency and health and safety. Depending on the type of work and the duration of exposure during working day, one or both arms can be exposed to vibration which may be transmitted to the sholuder, causing the various diseases that affect the blood vessels, nerves, bones, joints, muscles or connective tissues of the hand. Vibrations of machines and tools transmitted to a human body are recognized as a physical agents which could affect on worker's health and safety. So, the European Parliament and the Council addopted Directive 2002/44/EC which lays down minimum requirements for the protection of workers from risks to their health and safety arising or likely to arise from exposure to mechanical vibration. The Directive defines that daily vibration exposure limit value standardised to an eight-hour reference period shall be 5 (m/s^2) and the daily exposure action value standardised to an eighthour reference period shall be 2,5 (m/s^2) [1]. Based on both practical experience and laboratory experimentation concerning human response to hand transmitted vibration, the international standard ISO 5349-1 prescribes general requirements for measuring and evaluating hand transmitted vibration exposure, while ISO 5349-2 provides practical guidelines to perform measurement correctly.

2. CHARACTERIZATION OF HAND-TRANSMITTED VIBRATION EXPOSURE

The assessment of the level of exposure to hand-arm vibration is based on the parameter named daily vibration exposure, $a_{hv(eq,8h)}$, expressed in (m/s^2), [1,2]. It is the vibration total value, a_{hv}, normalised to an eight-hour reference period, i.e. 8-h energy-equivalent total value. Instead term $a_{hv(eq,8h)}$, a convinient alternative denotation is *A(8)*. The daily vibration exposure is defined by equation

$$A(8) = a_{hv(eq,8h)} = a_{hv}\sqrt{\frac{T}{T_0}} \qquad (1)$$

where T is the total daily exposure duration to vibration a_{hv}, and T_0 is the reference duration of working day (8 h or 28800 s).

Vibration total value, a_{hv} (m/s^2), is calculated from the three single-axes root-mean-square (r.m.s.) values of the frequency-weighted hand-transmitted vibration, a_{hwx}, a_{hwy} and a_{hwz}, according to equation

$$a_{hv} = \sqrt{a_{hwx}^2 + a_{hwy}^2 + a_{hwz}^2}\ . \qquad (2)$$

If the work process is such that the daily exposure consist of several operations with different vibration magnitude, than the daily vibration exposure A(8) should be obtain using equation

$$A(8) = \sqrt{\frac{1}{T_0}\sum_{i=1}^{n} a_{hvi}^2 T_i} \qquad (3)$$

where a_{hvi} is the vibration total value for *i-th* operation; n is the number of individual vibration exposures (operations); T_i is the duration of the *i-th* operation; T_0 is the reference duration of working day (8 h or 28800 s).

Since the vibration exposure is dependent of the magnitude of vibration and on the duration of the exposure, two main quantities should be evaluated for each *i-th* operation during exposure to vibration, that are the vibration total value a_{hvi} and the duration (per day) T_i of vibration exposure for operation *i*.

3. MEASUREMENT METHODOLOGY

Stages in which the evaluation of vibration exposure can be broken up are as follows [3]:

- Preparation of the mesurement procedure which include identification of discrete operations which make up the worker’s normal working pattern; selection of operation to be measured, estimation of daily vibration exposure duration, organization and duration of the measurement.
- Measurement of vibration magnitude, which implies selection of measurement equipment, location and attaching of accelerometers, check and verification of the measurement chain.
- Calculation of the daily vibration exposure and evaluation of uncertainties.

3.1 Preparation of the measurement procedure

The work process at the workplace may be composed of different operations, often repeating. As the operator can change tools during different working operations or change working regime during one operation, the level of vibration exposure may vary greatly from one operation to another. So, it is very important to select operations to be measured and choose an appropriate measurement method for each operation.

3.1.1 Selection of operation to be measured

To obtain a good picture of the worker’s average daily vibration exposure it is necessary to identify all machines and tools being used during working day, i.e. sources of vibration exposure. Further, all working modes of one power tool have to be identified. For example, chain saws may be idling, operating under load while cutting through a tree trunk, or operating under low load while cutting side branches. Also, it is good to recognize changes in the operating conditions which might affect vibration exposure. A good example is a grinder used initialy for bulk metal removal, then followed by more complicated operations of cleaning and shaping. It is allways useful to get information from workers and their supervisors about situations producing the highest vibration magnitude, as well as to use information from manufacturers on vibration emission values.

3.1.2 Organization of the measurement

The measurement can be organized in following ways:

a) Long-term measurement of continuous tool operation

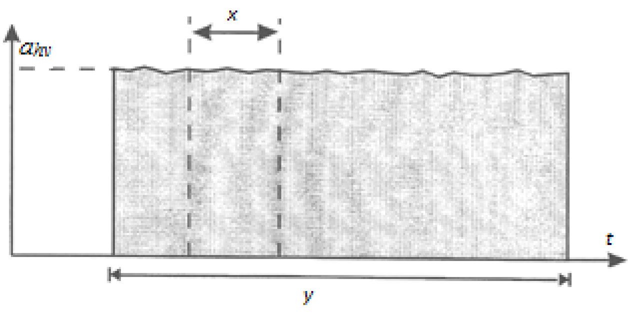

Fig. 1 *Long-term measurement of continuous exposur*e [3]

A good examples of this type of operation are floor polishing or ride-on lawn mowers. The worker's hand is always in contact with the power tool or hand-held workpiece during long-term continuous operation. The vibration magnitude can be measured over long periods and the exposure time is the time for which the power tool is in use. The coresponding vibration pattern is shown in Figure 1, where *x* represents duration of measurement and *y* represents operating time equal to exposure time in this case.

b) Long-term measurement of intermittent tool operation

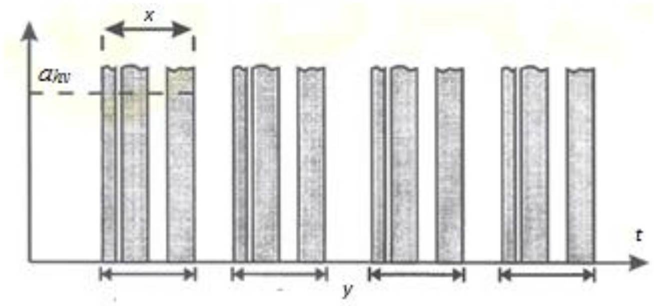

Fig. 2 *Long-term measurement of intermittent exposure* [3]

Examples of this type of operation are the use of grinders, chain saws and scaling hammers. The worker's hand is always in contact with the power tool or hand-held workpiece but the tool is not operated continuously because of short breaks in operation. In this case, the vibration magnitude should be long-term measured over a representative period of tool use (including periods when tool does not operated) and the exposure time is the time for which the tool is used during the working day. The coresponding vibration pattern is shown in Figure 2, where *x* represents duration of measurement and $y = \sum x$ represents total tool use time equal to exposure time in this case. The adventage of this type of measurement is that the vibration magnitude represents the actual task, including running-up and running-down periods of the machine, but at the other hand, the vibration magnitude is highly dependent on time the power tool is truly operating in worker's hand.

c) Short-term measurement of intermittent tool operation

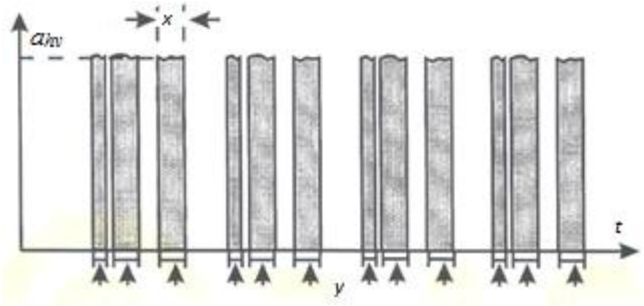

Fig. 3 *Short-term measurement of intermittent exposure* [3]

Examples of this type of operation are the use of hand-held grinders, pedestal grinders, chain saws and scaling hammers. The worker's hand is always in contact with the power tool or hand-held workpiece but the tool is not operated continuously because of long breaks in operation or the hand is taken off the tool during use. In this case, the vibration magnitude should be short-term measured over a period of continuous operation and the exposure time is the time for which the power tool is being operated during the working day. The coresponding vibration pattern is shown in Figure 3, where x represents duration of measurement and $y = \sum x$ represents total operating time equal to exposure time in this case. As the vibration measuremnet does not include periods when the machine is running-up and running-down to idling or off, this coud be considered as disandvatages of this type of measurement if run-up or run-down times are comparable to the time spent at operating speed.

d) Fixed-duration measurement of single impacts or bursts of tool operation

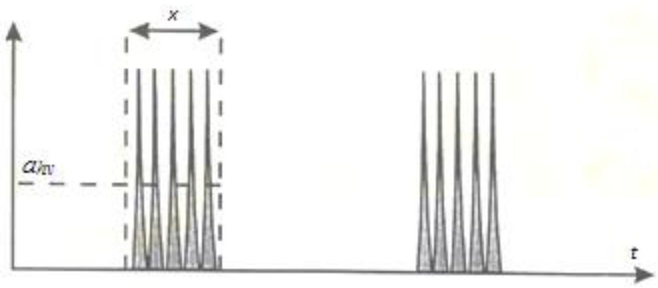

Fig. 4 *Fixed-duration measurement of single impacts or bursts of tool operation* [3]

Tipical examples of this type of operation are use of nail guns and powered impact wrenches producing irregular single impacts or bursts of vibration with long brekas between them. The average vibration magnitude should be measured over a fixed duration which include a known number (one or more) of impacts or bursts. The tipical vibration pattern is shown in Figure 4. The exposure time is calculated as the measurement duration multiplied by the number of impacts per day devided by the number of impact or bursts in the measurement period, according to

$$T = \frac{number\ of\ impacts\ per\ day}{number\ of\ impacts\ in\ measurement\ period} \times measurement\ duration \tag{4}$$

e) The vibration evaluation where more then one tool is used: If the worker uses more than one power tool during working day, then appropriate methods described above should be used to determine a partial vibration exposure $A_i(8)$ for each individual task. After each task was analysed separately, the total daily vibration exposure can be calculated according to

$$A(8) = \sqrt{\sum_{i=1}^{n} A_i^2(8)} \tag{5}$$

3.1.3 Duration of vibration measurement

It is necessary to average the measurement over a representative period for typical use of a tool or process. It is recommended to start the measurement at the moment when worker puts his hands to vibrating surface and finish it when contact between hands and tool is terminated. During measurement, a variations in the vibration magnitude are possible, as well as periods with no vibration exposure.

The minimum acceptable duration of measurements depends on the vibration signal, used instrumentation and operation characteristics. The total measuring time (that is the number of samples multiplied by the duration per measurement) should be at least 1 minute. It is preferable to take a number of shorter duration measurement samples than to perform a single long duration measurement.

3.1.4 Estimation of daily vibration duration

It is necessary to determine the daily exposure duration for each vibration source. It could be done by measuring the concrete exposure time during a period of normal use of vibration source, by use of stopwatch or analysis of the work process's video recording (it is the most reliable source of information).

Another way to determin the daily exposure duration is to get information on work rate (for example, the number of work cycles per shift or the shift length).

If the vibration has been averaged over a complete work cycle, then the duration of the work cycle multiplied by the number of cycles per day defines the daily exposure time.

3.2 Measurement of vibration magnitude

3.2.1 Measuring equipment for hand-transmitted vibration

Vibration measurement should be performed by equipment confirming to the requirements of ISO 8041 [3]. It is necessary to check this instrumentation for correct operation before and after use.

Basic measurement system consists of vibration meter unit and accelerometer. The single-unit vibration meter must have built-in frequency weightings and integrating facilities. There are also more sofisticated measurement systems based on frequency analysis (e.g. one-third-octave or narrow band), which may use digital data recorders to store amplitude-time information, or use computer-based data acquisition and analysis techniques.

The type of accelerometer to be chosed for measurement depends on the expected vibration magnitude, the required frequency range, the physical characteristics of the surface being measured and enviromental conditions.

3.2.2 Location and orientation of accelerometers

It is prefferable to perform simulataneous triaxial measurement of vibration using the triaxial accelerometer. In practis, the basicentric coordinate system is used, with the origin in a vibrating appliance, workpiece, handle or control device gripped by the hand, as described in ISO 8727.

Vibration sholud be measured at or near the surface of the hand(s) where the vibrations enters the body, preferably at the middle of the gripping zone. Thus, it is important before acctual measurement to observe how a power tool or hand-held workpiece is held and to indentify the best location and orientation of the accelerometer. Figure 5 shows examples of measurement locations for some common power tools.

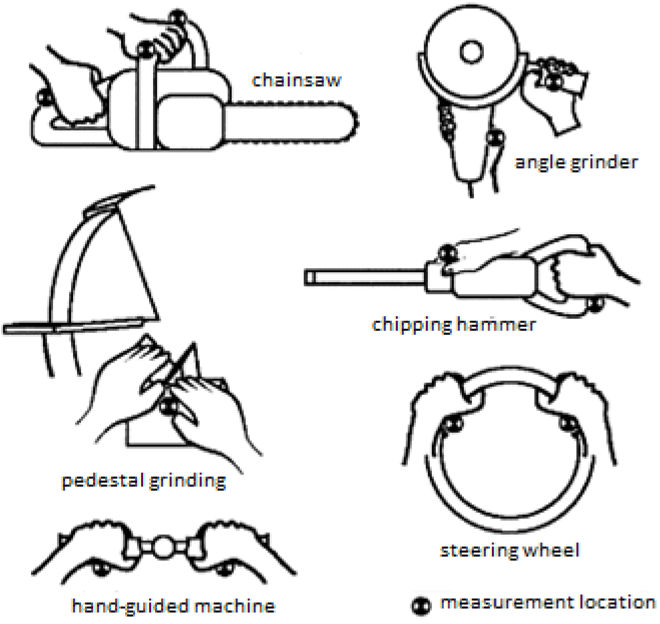

Fig. 5 *Examples of measurement location* [3,4]

Additionally, International Standars ISO 8662-2 to ISO 8662-14, ISO 7505 and ISO 7916 specify measurement locations for measuring the vibration at the handles of different hand-held power tools for the purpose of determining vibration emission values.

For avoiding interference with the power tool controls or with the safe operation, the best location is where the on-off switch is positioned. It is often necessary to use special mounting adaptors for transducer which should fit under the hand or between fingers.

3.2.3 Attaching accelerometers

The accelerometers should be rigidly attached to the vibrating surface. Typical mounting methos, some of these depiched in Figure 6, are follows:

- stud mounting (screwed),
- mounting by glue or cement,
- clamp or nylon strap connections,
- hand-held adaptors.

The choise of mounting method depends on the particular measurement situation and each method has some advantages and disadvantages. The mounting system sholud have a flat frequency response across the range of frequancies being measured, i.e. it should not attenuate or amplify and should not have any resonances in the frequency range. It is inevitably that mounting of an accelerometer causes some disruption in work operations. Thus, mounting sholud be aranged in that way the operator can work as normaly as possible.

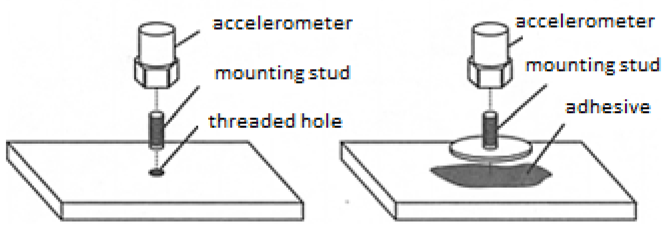

Fig. 6 *Examples of stud mounting method* [3,4]: *accelerometer can be mounted by a stud screwed into a threaded hole or adhered to the tool by glue or wax*

3.2.4 Check of measurement chain

The whole measurement chain sholud be checked before and after a measurement, by using a vibration calibrator as a reference vibration source producing known sinusoidal acceleration at a known frequency.

3.3 Calculation of the daily vibration exposure

The daily vibration exposure *A(8)* in (m/s^2) is calculated according to equation (1). In order to compare different operations and to evaluate the individual contribution of a particulare operation to the daily exposure *A(8)*, it may be useful to calculate the partial vibration exposure for the individual operation, $A_i(8)$, using

$$A_i(8) = a_{hvi}\sqrt{\frac{T_i}{T_0}} \tag{6}$$

where a_{hvi} is the vibration total value for *i-th* operation; T_i is the duration of the *i-th* operation; T_0 is the reference duration of working day (8 h or 28800 s).

The daily exposure is then given by equation

$$A(8) = \sqrt{\sum_{i=1}^{n} A_i^2(8)} \tag{7}$$

where n is the number of operations.

3.3.1 Uncertainties of evaluation of daily vibration exposure

The uncertainties associated with the evaluation of *A(8)* are often high (20-40%), so it is normally to present the value of *A(8)* with two significiant figures.

The uncertaintes associated with instrumentation and calibraton, electrical interference, mounting and mass of the accelerometer, are usually small compared with those resulting from selection of measurement location and variability of the work operation.

The uncertainties of the estimation of exposure time resulted from uncertainties in measurement of duration of exposure, wrong estimates of the number of work cycles per day or wrong information supplied from workers.

The experimenter sholud determine the main sorces of uncertainties and multiple measurement should be made in order to determine the extend of uncertainties and to calculate the standard deviation regarding the dominant sources of uncertainty.

4. CASE STADY

The exposure of workers to the hand-transmitted vibrations is especially present in those industries in which the production process is based on the use of hand-held power tools and machines, such as pneumatic, electric, hydraulic or internal combustion engine-driven chain saw, pecussive tools or grinders. A good example of such an industry is the “DelCin” company from Banja Luka engaged in the production of metal furniture. In order to assess the risk of exposure of workers to harmful effects of mechanical vibration, the measurement of hand-transmitted vibration was carried out for a few operations in the department for grinding and polishing.

4.1 Preparation of measurement

Preparation of measurement includes selection of operations to be measured, decision making about the measurement organisation and estimation of the vibration exposure duration.

Since each worker uses one power tool during whole working day, it was decided to analyze the vibration exposure of workers at workplaces where following power tools are used:

1. Random orbital sander
2. Small angle grinder
3. Angle sander
4. Pedestal belt grinder

The workpiece is the metal construction of chair built from 8 rods of rectangular cross section. Since each rod has 4 perperdincular surfaces, that is 32 surfaces in total to be machined. By colecting some information from workers, the examiner concluded that an average of 50 chairs are worked on per day. Also, by oberving the work proces and measuring duration of some operation, the examiner found that an avreage duration of one operation sequence (one surface of the regtangular rod) is about 13 s. Thus, for each chair, the operating cycle is made up of two main periods:

- 415 s to grind each of 32 surfaces of the workpiece;
- 35 s to manipulate with the workpiece (rotation of the workpiece and replacement of sandpaper, during which the tool doesn’t operate).

The vibration pattern is similar to that shown in Figure 2. It was decided to do the measuremet as a long-term measurement of intermittent tool operation. One operating cycle has a duration of 450 s. At a work rate of 50 workpieces per day, the total daily exposure time is then 375 min, that is 6,25 h.

4.2 Measurement of the vibration magnitude

For this analysis, the following equipment, showed in Figure 7, was used:

- hand-held analyzer type 4447,
- triaxial accelerometer type 4524 B 001,
- adaptors for fixing and locating the accelerometer to the gripping zone of vibration, all produced by Bruel&Kjaer.

Fig. 7 *Hand-held vibration analyzer, accelerometer and adaptors (L-shape, T-shape, cube)*

4.2.1 Mounting the accelerometer

Regarding to specifics of used power tools, three different adaptors were used in this analysis. In the case of the pneumatic random orbital sander, the worker holds the T-shape hand adaptor equiped with accelerometer between his fingers, in the midle of the gripping zone, Figure 8.

Fig. 8 *Random orbital sander: the accelerometer attacheb by T-shape hand adaptor*

When the small angle grinder is used, the worker holds the L-shape handle adaptor equiped with accelerometer on the underside of the tool handle, Figure 9.

Fig. 9 *Small angle grinder: the accelerometer attached by L-shape hand adaptor*

In the case of the angle sander, the accelerometer was fixed by the cube adaptor, Figure 10.

Fig. 10 *Angle sander: the accelerometer attached by nylon strap*

The specificity of the operation done on the pedestal belt grinder is that the worker holds the workpiece in his hands and feeds workpiece to the working part of the machine, such that the workpiece is the source of vibration transmitted to the hands, Figure 11a. Because of that, the accelerometer fitted in the cube adaptor was mounted directly to the metal rod of the chair, Figure 11b.

Fig. 11 *Pedestal belt grinder: the accelerometer attached by nylon strap*

4.2.2 Measurement results

The primary quantity which is measured and used to describe the vibration magnitude is the root mean square (r.m.s.) frequency-weighted acceleratinon for the x, y and z axis, a_{hwx}, a_{hwy} and a_{hwz}. The evaluation of vibration exposure is based on the vibration total value, , which combines all three axes, according to equation (2). All quantities are expresed in (m/s^2).

All measured r.m.s. values and vibration total values for each investigated workplace are shown in Table 1.

Table 1 *Results of measurement*

Vibration values (m/s^2)				
Work place	1	2	3	4
a_{hwx}	2,7	1,4	2,7	1,1
a_{hwy}	5,8	2,2	5,8	0,8
a_{hwz}	3,1	0,5	1,9	0,8
a_{hv}	7,2	2,7	6,7	1,6

4.3 Evaluation of the daily vibration exposure

Daily vibration exposure is quantityderived from the magnitude of vibration (vibration total value) and the daily exposure duration, accorting to the equation (1).

Since the duration of operatig cycle for each workplace is about 450 s, and there are 50 cycles per working day, the daily exposure time is

$$T = \sum_{i=1}^{50} t_i = 450 \times 50 = 22500s\ (6,25h) \qquad (8)$$

The daily vibration exposure A(8), calculated by equation (1), for each workplace is

- Random orbit sander

$$A(8) = a_{hv}\sqrt{\frac{T}{T_0}} = 7,2\sqrt{\frac{6,25}{8}} = 6,4 \text{ (m/s}^2\text{)} \qquad (9)$$

- Small angle grinder

$$A(8) = a_{hv}\sqrt{\frac{T}{T_0}} = 2,7\sqrt{\frac{6,25}{8}} = 2,4 \text{ (m/s}^2\text{)} \qquad (10)$$

- Angle sander

$$A(8) = a_{hv}\sqrt{\frac{T}{T_0}} = 6,7\sqrt{\frac{6,25}{8}} = 5,9 \text{ (m/s}^2\text{)} \qquad (11)$$

- Pedestal belt grinder

$$A(8) = a_{hv}\sqrt{\frac{T}{T_0}} = 1,6\sqrt{\frac{6,25}{8}} = 1,4 \text{ (m/s}^2\text{)} \qquad (12)$$

Results of analysis shows that the workers using random orbit sander and angle sander are exposed to vibration which daily exposure values exceeds the exposure limit value of 5 (m/s^2) prescribed by the EU Directive 2002/44/EC. Therefore, it is necessary the eployer takes some measures to protect workers from the risk to their health and safety arising or likely to arise from exposure to mechanical vibration.

5. CONCLUSION

After completion the evaluation of exposure of workeres to the hand-transmitted vibration and once the exposure limit value is exceeded, the employer has an obligation to establish and implement a programme of technical and organisation measures intended to reduce vibration exposure to a minimum value. A good technical preventive measure which may be implemented immediately is reorganization of working process in a way that one worker uses more than one power tool, i.e. to performs more then one operation by use of different tools. In this way, worker may use the power tool producing too high vibration magnitude for a shorter period. In combination with the other tool producing low vibration magnitude, it is possible to reduce the overall daily vibration exposure. Apart from this, if there is a choise between different tools, the tool resulting in the lowest vibration exposure should be used. All equipment should be carefuly maintained. Also, tools with handle shapes which results in high pressures on the skin in the area of contact should be avoided. It is always preferable to use tools requiring the smallest contact grip and feed forces. The mass of hand-held tools sholud be kept to a minimum, provided that vibration magnitude or contact forces are not increased. Anti-vibration gloves can be useful where it is shown that use of gloves reduce the vibration exposure.

REFERENCES

[1] Directive 2002/44/EC of the European Parliament and the Council on the minimum health and safety requirements regarding the exposure of workers to the risks arising or likely to arise from physical agents (vibrations), 2002.

[2] BAS EN ISO 5349-1:2001, *Mechanical vibration – Measurement and evaluation of human exposure to hand-transmitted viration – Part 1: General requirem*ents.

[3] BAS EN ISO 5349-2:2001, *Mechanical vibration – Measurement and evaluation of human exposure to hand-transmitted viration – Part 2: Practical guidance for measurement at the workplac*e.

[4] N. J. Mansfield, *Human Response to Vibration*, CRC Press, 2005.

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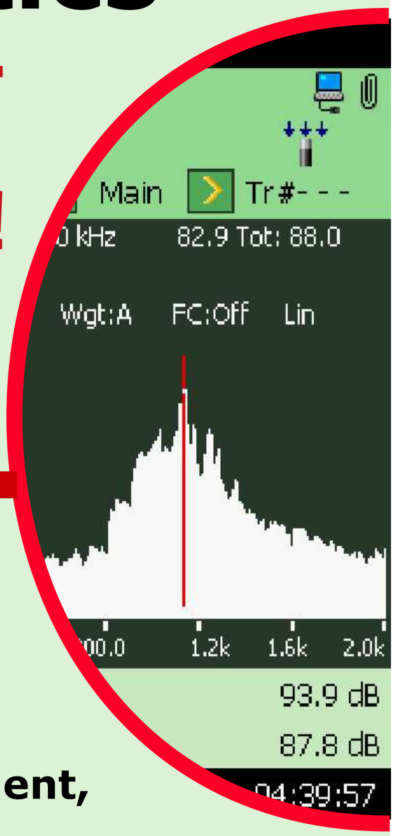

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ENGINEERING OF IN-HOUSE DYNAMIC LOAD CELL

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Abstract - *Load cells transducers are used to measure an applied force as an electrical signal with magnitude proportional to that force.*

Strain load cell sensors are suitable for accurate dynamic and static measurement. Designed as layer of fine grade wire or foil that is bonded to a carrier element, the variance of the measured electrical resistance is proportional with the variance of strain. The Wheatstone Bridge Circuit used in static strain measurement for its outstanding sensitivity can be adapted for dynamic measurements of low displacement. Although commercially available load cells are cast of stainless steel, depending on the material used and its geometry a wide range of measurements can be achieved.

This papers presents the stages for the development of strain gauge load cells within the framework of the Mechanics and strength of materials department, Laboratory of vibrations, Politehnica University Timisoara.

1. INTRODUCTION

Load cells are transduces used to measure force or applied loads through deformation (strain). The applications of load cells are so common, that they are used almost everywhere a force needs to be measured. Although there are various types of load cells (hydraulic, pneumatic), strain gauge cells are the most common due to their stiffness and long life cycle.

Strain gauge load cells work on the principle that the strain gauge (a planar resistor) deforms/stretches/contracts when the material of the load cells deforms appropriately. These values are extremely small and are relational to the stress and/or strain that the material load cell is undergoing at the time. The change in resistance of the strain gauge provides an electrical value change that is calibrated to the load placed on the load cell [1].

Load cells usually consists of four strain gauges in a Wheatstone bridge configuration – full bridge, but cells with two strain gauges - half bridge or even one strain gauge - quarter bridge are available depending on application and its setup. Strain gauge load cells are passive devices and the output signal is typically in the order of a few millivolts, often microvolts, and require amplification by an instrumentation amplifier before it can be of any use [2].

There are quite several shapes of load cells forms due to the particularization of each measurement type, but the most common could be divided into: pancake load cells, S beam load cells, load button and single point load cells.

The S Beam Load Cell, also known as the Z Beam Load Cell, is one of the most popular types of load cells due to their versatility in measuring. Its high precision, low price, and ease of installation makes it very popular. Non the less, caution must be taken because S beam load cells are strictly designed for in-line applications, and as such they are very sensitive to extraneous load, torque, and moments [3].

Performance parameters are likely to be the primary considerations in the purchase of a load cell. After all, if the cell does not perform as the design demands, it is worthless to the test. Depending on the application, any of the following factors may be critical to the selection process: accuracy, compensated temperature, creep, deflection, hysteresis, natural frequency, non-linearity, non-repeatability, operating temperature, rated load, rated output, resolution, safe overload, temperature shift, and zero return.

The accuracy, or tolerance of average deviation of the actual output from theoretical output, will be an important factor in applications where there is little margin for error in measurements.

Development of such devices in house, as difficult as it may seem, is possible and it is presented in the following pages.

2. GEOMETRIC AND MATERIAL CONSIDERATIONS

When a force is applied to the load cell, the beam (or bridge) onto which the strain gauge is mounted, is subjected to combined bending and axial compressive stresses. The bending stress is by far the dominant component, hence the term "bending beam". Importantly, the relationship between the applied force and the combined stress in the beam is linear. An important feature of the load cell is that total displacement under load is very small. This is significant, as it nearly eliminates unwanted dynamic effects associated with mass. Choosing the S beam type is validated by the desire to maintain colinearity through the mounting points (Figure 1).

Generally, a load cell will be designed based on available material, to minimize the amount of metal cutting. For example, the design might be based on an available bar of aluminum, which needs only to be cut to the required length. Therefore, it is only necessary to drill the body hole, cut the slot, drill & tap the mounting hole. Particular care should be taken with regard to the body hole. The location should be accurately marked out, then drilling should be done by drilling a series of progressively larger holes, to ensure that the location does not "drift" while drilling. The bottom mounting hole should be drilled at approximately mid-width, then tapped to the desired thread size. The slot is simply cut with an angular grinder.

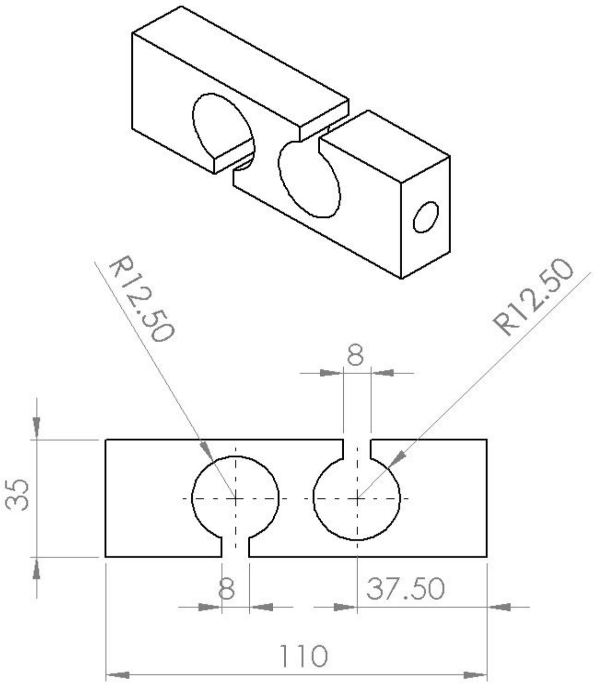

Figure 1- *Geometric dimensioning*

A yield strength or yield point is the material property defined as the stress at which a material begins to deform plastically. Prior to the yield point the material will deform elastically and will return to its original shape when the applied stress is removed. Once the yield point is passed, some fraction of the deformation will be permanent and non-reversible, meaning the load cell is not reusable. The yield point determines the limits of performance for mechanical components, since it represents the upper limit to forces that can be applied without permanent deformation.

Through iterative operations, desired geometric dimensions with proportional dependency between gauge and tested items can be determined [4]. Selecting special alloys to manufacture load cells is not a critical condition. As the range of measurement for our load cell was between 50 and 1000 N, any number of materials can be chosen for the designed shape as long as its yield strength is in the linear region of the stress-strain diagram (Figure 2).

Von Mises stress is considered to be one of the most reliable methods for design engineers to test their products. Using this information an engineer can say his design will fail, if the maximum value of Von Mises stress induced in the material is more than strength of the material. It works well for most cases, especially when the material is ductile in nature. According to this theory, failure occurs when the distortion energy in actual case is more than the distortion energy in a simple tension case at the time of failure [5].

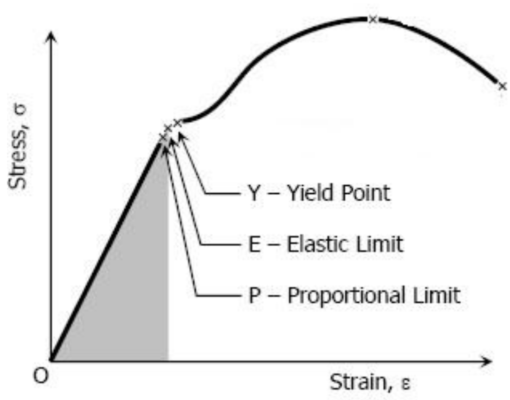

Figure 2 - *Generic steel stress-strain correlation*

With the aid of finite element analysis , it was determined that for our chosen geometry, even one of most common types of steel, S235JR, is useable [6]. The maximum equivalent stress was 1.7674e+008 Pa for a chosen force of 1000N (Figure 3). This value is way below any general purpose steel's tensile strength limit (350MPa)[7].

Having the maximum equivalent stress below the proportionality limit, keeping in mind that stress is proportional to strain (Hooke's law), the stress-strain graph is a straight line, and the gradient will be equal to the elastic modulus of the material 200GPa.

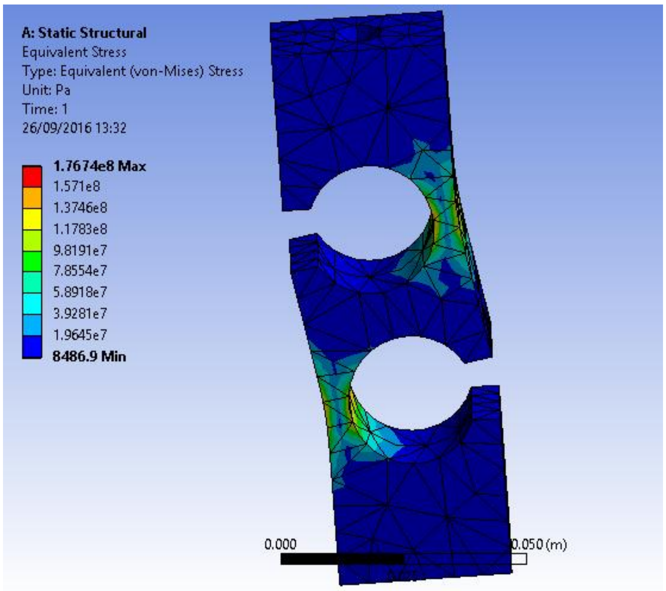

Figure 3 - *FEM equivalent stress*

Yield strength testing involves taking a small sample with a fixed cross-section area, and then pulling it with a controlled, gradually increasing force until the sample changes shape or breaks. During the tensile test, the longitudinal strain was recorded using mechanical extensometers. For this purpose, the Instron 8872 fatigue testing system was used to determine the properties of the chosen material, resulting in data that fits the S235JR type of steel [8].

3. STRAIN GAUGE TESTING

A strain gauge is a device that measures electrical resistance changes in response to, and proportional of, strain applied to the device. The most common strain gauge is made up of very fine wire, or foil, set up in a grid pattern in such a way that there is a linear change in electrical resistance when strain is applied in one specific direction, most commonly found with a base resistance of 120Ω, 350Ω, and 1000Ω [9].

Each strain gauge has a different sensitivity to strain, which is expressed quantitatively as the gauge factor (GF). The gauge factor is defined as the ratio of fractional change in electrical resistance to the fractional change in length (strain) [10].

The gauge factor for metallic strain gauges is typically around 2, our strain gauges having a factor of 2.11.

Strain measurements rarely involve quantities larger than a few millistrain. In our case, to bypass most impediments, we used a Vishay P3 Strain Indicator. The Model P3 Strain Indicator and Recorder is a portable, battery-operated instrument capable of simultaneously accepting four inputs from quarter-, half-, and full bridge strain-gage circuits, including strain-gage-based transducers. Designed for use in a wide variety of physical test and measurement applications, the P3 functions as bridge amplifier, static strain indicator, and digital data logger (figure 4).

Standard sensor input connection is via eccentric-lever release terminal blocks. Optional transducer connection is available via side-mounted bayonet locking circular connectors.

Data, recorded at a user-selectable rate of up to 1 reading per channel per second, is stored on a removable flash card and is transferred by USB to a host computer for subsequent storage, reduction and presentation with the supplied software is ideal for specific load/unload cycles (figure 5).

A highly stable measurement circuit, regulated bridge excitation supply, and precisely settable gage factor enable measurements of ±0.1% accuracy and 1 microstrain resolution. Bridge completion resistors of 120 ohms are built in for quarter-bridge operation.

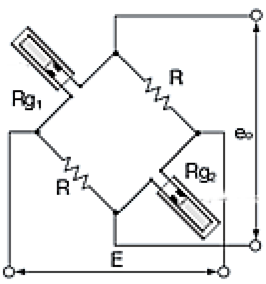

Figure 4 - *Strain gauge setup*

The calibration of the prototype load cell was done on a Multitest 5i tensile & compression test machine with a 5kN force cell [11].

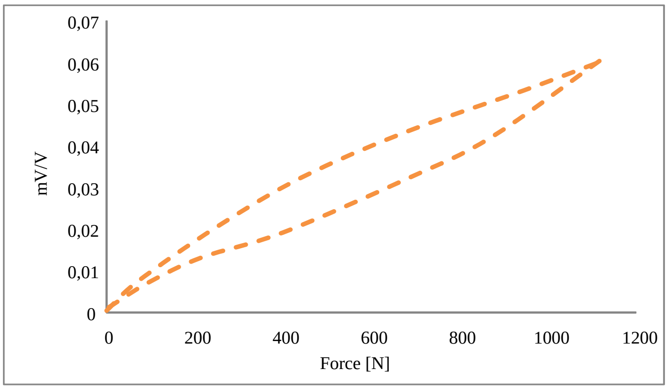

Figure 5 - *Load cell hysteresis*

Hysteresis is measured by increasing the load until it reaches some arbitrary value, then starting again with values around our maximum load value of 1000N and decreasing it until it reaches the same arbitrary value used before. The hysteresis is then the difference between the measured output when the load is increased to that value and that when the load is decreased to that value. Essentially, this being the difference between load cell output readings for the same applied load, one reading obtained by increasing the load from minimum load and the other by decreasing the load from maximum load.

The operating temperature, is usually important as long as the operating conditions of the load cell will be within this range. ISO 17025, Section 5.3, requires the control, monitoring and recording of laboratory environmental conditions as required by relevant specifications, methods and procedures, or where they influence the quality of the results. In our case we monitor temperature and humidity with AM2302 sensor. Throughout the year the only large variation being at the beginning of the cold season.

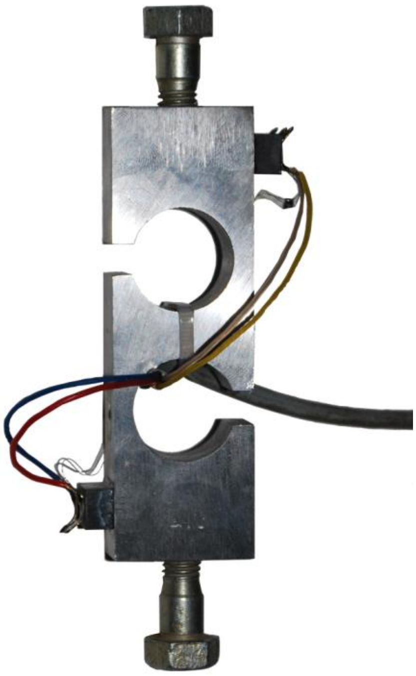

Figure 6 - *In house made load cell*

CONCLUSIONS AND RESULTS

The load cell's response time is an essential factor to consider for a dynamic application. The typical load cell behaves like a stiff spring that oscillates, so to achieve an accurate reading, the load cell must settle - that is, stop oscillating - in less time than the required weighing period. While load cell response time is typically not important for static application, a relatively high-speed testing rig requires fast-responding load cells. Such load cells dampen their own natural oscillating frequency when a load is applied to them. For sample rates less than 10Hz, the output of the load cell measured by the Vishay P3 Strain Indicator and Recorder and USB1208FS analog input acquisition device from Measurement Computing compared with the calibrated Instron 8800 series fatigue testing machine load values have an offset up to 2 %.

Creep factor of 0.2 %/min was detected at constant loads above 900N in the first 2 minutes. with all environmental conditions and other variables remaining almost constant.

Non-repeatability, as difference between load cell output readings for repeated loadings under identical loading conditions is a little bit high ±0.04 % over the load cell's full range.

Temperature effect on load cell was not tested. Having both strain gauges on opposite side of the Wheatstone bridge, they would compensate for minor temperature variation. Considering the working condition around 20-24 degrees C and humidity around 50 %, according to the strain gauge manufacturer, adhesive used for fixing, reliable data can be read for at least 6 months before the cell would need recalibration.

The outline of the load cell although seemingly simple, does pose challenges in developing and after careful calculus, design and simulation, resulted a model that is relatively simple to manufacture with conventional technology.

Development of such a device can be done in relatively short time, but its quality and responsiveness is proportional to the effort used. Although manufacturing of the load cell, excepting down times, was less than 4 man-hours, the engineering, calculus and simulations to choose geometry and material took over 20 man-hours. It could be said that a load cell was built from scratch in 1 day. The problem is that this load cell is useless without knowing its responsiveness for different loads and testing conditions. Testing the load cell, calibrating, comparing its response with other load cell specifications was an exhausting job. In our opinion, a load cell can be made by anyone with quite good resolution, but for high precision measurements it is far more convenient to buy a series produced load cell, even though its price would be 10-20 or even more times expensive.

REFERENCES

[1] Strain gauge HBM, https://www.hbm.com/en/0014/strain-gauges/ 08.06.2016

[2] J.F.Doyle, Modern Experimental Stress Analysis: Completing the Solution of Partially Specified Problems-International Edition,Wiley, ISBN-13: 978-8126539857, 2012.

[3] Woodward, "*Novel load cell for measuring axial force, shear force, and bending movement in large-scale structural experiments*", Experimental Mechanics, Volume 42, Issue 1, pp 115–122, 2002.

[4] M.S.M. Ghanvat, H.G. Patil, "Shape Optimization of 'S' Type Load Cell Using Finite Element Method", International Journal of Engineering Innovation & Research, Volume 1, Issue 3, pp 310-316, 2012.

[5] ANSYS Inc., Theory Reference for release 15.0. ANSYS User's Manual, http://148.204.81.206/Ansys/readme.html 12.06.2016

[6] D. Blakeborough. M.S. Clément,. Williams,. N. A. Kamble Vijay, P. N. Gore, "*Use of FEM and Photo Elasticity for Shape Optimization of S Type Load Cell"*, Indian Journal of Science and Technology, Vol. 5 No. 3, pp 2384-2390, 2012.

[7] I. Dumitru et al., "*Considerations on the Necessity of Determining Principal Stresses and Directions for Analysis of Material Durability under Multiaxial Fatigue*", 14th Symposium on Experimental Stress Analysis and Materials Testing Location: POLITEHNICA Univ Timisoara, Timisoara, ROMANIA Date: MAY 23-25, 2013

[8] Instron 8872 fatigue testing system http://www.instron.us/en-us/products/testing-systems/dynamic-and-fatigue-systems/servohydraulic-fatigue/8872-table-top 11.12.2008

[9] NI Measuring strain with strain gauges, http://www.ni.com/white-paper/3642/en/, 11.05.2016

[10] Futek Advanced Sensor Technology, Load Categories. http://www.futek.com/product.aspx?t=load&cat=sb 03.05.2016

[11] Multitest 5i http://www.mecmesin.com/multitest-5-i-5kn 21.08.2016

MODELING AND IDENTIFICATION OF VEHICLE VIBRATION

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Abstract *- Vibration of road vehicle affect the comfort passengers, vehicle life, road load and traffic safety. The sources of these vibration are the road surface roughness, the aerodynamical disturbances and engine working processes. In this paper the possibilities to modeling and identification of vehicle random vibration are considered. An experimental system is designed and used. The experiments are carried out in real driving condition at different speeds. The digital signal processing methods based on the power spectral density analysis to identification and interpretation of vehicle vibration are used. An appropriate model for vehicle vibration simulation is developed. Obtained experimental and simulation results are discussed.*

Key words**:** *random vibration, modeling, simulation, experiment, identification*.

1. INTRODUCTION

Most of the published results of investigation into the vehicle handling characteristics concern to simplified models, subjected to pure harmonic or transient excitation . But, in reality, the vehicle as complex non-linear dynamical system is subjected to random excitation from the environment as follows, 1) uneven road, 2/ changeable state of adhesion in contact, tyre-road, 3/ aerodynamical effects, 4/ effects of the driver control. Roughness of the road surface is the most important source excitation of the vehicle, and is a major cause of the discomfort of passengers [1], [2], [3]. On the other hand, fluctuation in the wheel loads, caused by the road surface, also conflict with requirements for safety movement. Namely, oscillatory processes arising from the movement of motor vehicles on the unevenness roadway, generate dynamic forces that affect to handling characteristics, ie., road holding and ride-comfort, as well as to life of the elements and components of vehicles but also lead to fatigue of the driver and passengers, which is reflected in their ability to work and health, [4], [5], [6],[7], [8], [9].

Bearing in mind the above presented problems, in this paper we discussed about some specific vehicle handling with respect to modeling, simulation, and experiments coduction.

2. METHODOLOGY

Based on the conducted analysis and conclusions given in the introduction, this chapter will present the subject of work and used methodology.

In order to highlight and solve the task, a physical model of the vehicle was created and shown in Figure 1. In general case, a two-axle vehicle with four wheels and four tracks is presented as a spatial physical model with five masses: sprung mass, M, and four unsprung masses, m_{11}, m_{12}, m_{21}, m_{22}, and a total of ten degrees of freedom, x, z , y, θ, ψ, ε, z_{11n}, z_{12n}, z_{21n}, z_{22n}. Characteristics of stiffness and damping of suspension are marked by c_{ij}, k_{ij}, i = 1,2, j = 1,2, respectively, and characteristics of stiffness and damping of the tire by c_{ijp}, k_{ijp}, i = 1,2, j = 1,2, respectively , [10], [11].

Potential properties of excitation effects of the environment on a vehicle are marked by vectors as follows: R - the uneven road, F - by changing the conditions of adhesion, A - from the air environment, V - from the effects of the driver control. In doing so, a distinction is made between the term of "potential properties of excitation effects" and the term "realized excitation interaction between the vehicles and the environment, ie. with a driver". A simple explanation for this is found in an example of excitation of vehicles from road roughness. The potential characteristics, in this case, are actual macro- and micro-roughness of the road, and the excitation resulting from the interaction between the road and the vehicle is carried out in the course of the movement, which is affected by numerous factors.

The model structure given in Fig. 1, can be simplifyed depending from defined research tasks and given condition.

For example, in the case of the vehicle longitudinal symmetry, basic spatial model with five masses and ten + one degrees of freedom can be reduced to basic model in vertical – longitudinal plane so – called “two – dimensional” model with three masses and four + one degrees of freedom. In the case of longitudinal and transversal vehicle symmetry, ie., by neglected the rolling and pitching motions, the basic spatial model, can be represented by four symmetrical quarter models concentrated masses system, so-called, “one-dimensional”, corresponding to four wheels subjected to vertical motion. With this assumption the oscillatory processes of the whole system are described by means of the oscillatory processes of individual basic quarter models, with two masses and two degrees of freedom.

Considering that the mass of the seat with the driver is significantly less than the total mass of the vehicle, then this mass exerts a negligible effect on overall levels of oscillating systems. This allows the system vehicle-seat-driver to decompose into two oscillatory models; the first is three- , or two- or one-dimensional model of the vehicle, according to above presented model structures, and the second is one sub-model, of the seats, with one concentrated mass, on the vehicle body, as a “mobile platform”. In this way, the vehicle model can be analysed separately from the model of the seats, and thus it is possible to describe its relevant oscillatory processes depending on influential parameters, then the same could be imported into a database and used in all cases of seat's selection or replacement. On the other hand, for the analysis of oscillatory processes of the seats, in terms of its properties, can be used partial sub-model, excited by oscillations of the moving platform, ie, vehicle's chassis, which can be quickly generated or used from database

For above described two – dimensional vehicle model are written differential equations in the condensed implicit form,

$$\ddot{z} = \ddot{z}(M, F_{ck1}, F_{ck2}) \tag{1}$$

$$\ddot{\alpha} = \ddot{\alpha}(I, a, b, F_{ck1}, F_{ck2}) \tag{2}$$

$$\ddot{z}_{11} = \ddot{z}_{11}(m_1, F_{ck1}, F_{pck1}) \tag{3}$$

$$\ddot{z}_{22} = \ddot{z}_{22}(m_2, F_{ck2}, F_{pck2}) \tag{4}$$

$$F_{pck1} = F_{h11}, \quad F_{pck2} = F_{h22} \tag{5}$$

$$F_0(v) = \sum_{i=1}^{6} R_i, \, i = f, v, j, p, \alpha \tag{6}$$

where is, z'' - vertical acceleration at center of vehicle sprung mass, α'' - angular acceleration about the transverse axis of the vehicle – z''_{11}, z''_{22} –vertical acceleration at front, rear wheel center, respectively, M – vehicle sprung mass, m_1, m_2 – vehicle unsprung masses, front, rear, respectively, I – moment of inertia about the transverse axis of the vehicle, F_{ck1}, F_{ck2} – resultant of elastic and damping forces front, rear suspension, respectively, F_{pck1}, F_{pck2} – resultant of elastic and damping forces front and rear tire, respectively, F_{h11}, F_{h22} – resultant of elastic and damping forces in contact tire – road, $F_0(v)$ – vehicle tractive force, R_i – components of vehicle motion resistances.

For presented quarter models differential equation can be written analogous equations from (1) to (6), without (2). Corresponding indices of concentrated masses, acceleration, forces must be adjusted to model structure and its parameters.

For experimental research two experimental systems were used according to presentation in Fig. 2 and Fig. 3. The first system, shown in Fig. 2, presented a universal purpose self driven measuring platform with equipment as support to simulation and identification studies. For the purposes this work this platform was used for preliminary testing and identification element's sub-models and components of the vehicle such as, sprungs, damper, tyres etc.

Details of the second experimental system, ie., experimental passenger car are presented in Fig.3, [5], [7], [11], as measuring places of vertical acceleration in front- , rear wheel centres, a/, b/, respectively and on the place of connection of front right damper on the vehicle body. For vertical acceleration measurement were used HBM B12 inductive sensors in connection with 8.chanel measuring and data acquisition system HBM Spider 8 and computer. One complete experimental vehicle system includes eight HBM acceleration sensors, four in wheel centres, and four on places of connection of shock absorbers and vehicle body, according to spatial model structure in Fig. 1, ie., presented levels I and II. For two–dimensional vehicle model number of measuring points is reduced to four, and for quarter model, a/ basic, 2, b/ with the seat, 3.

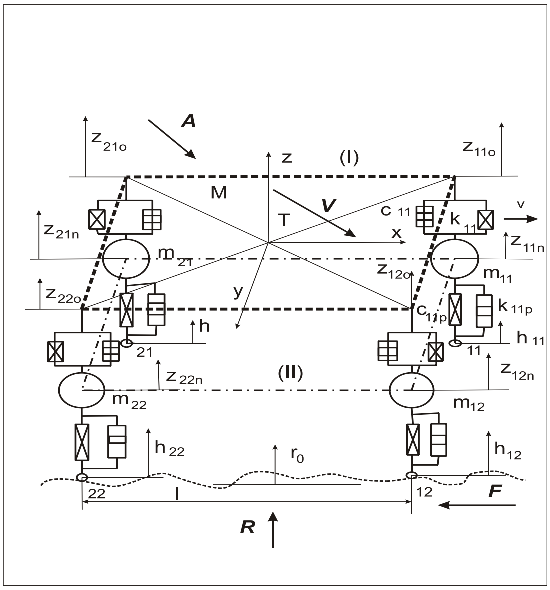


Fig. 1 *Vehicle simulation model*

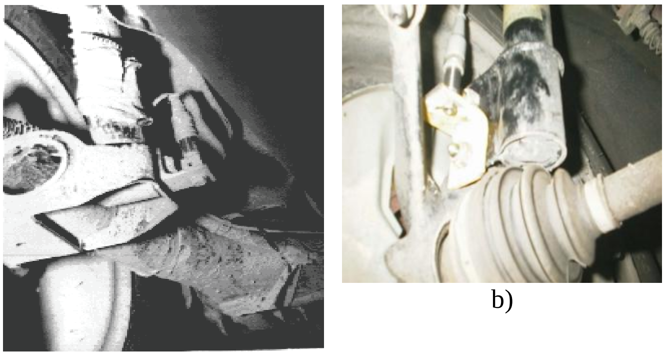
a)

b)

Fig. 2 *a), b) Self driven measuring platform with equipment*

3. RESULTS

Some illustrative experimental results of this study are shown in Fig. 4, 5 and 6. Simulation results are shown in Fig. 7 and 8, for quarter model and in Fig. 9, for two–dimensional (2D) model. Experiments were conducted in different road conditions and with different regimes of vehicle movement. Time record of vertical acceleration at centre of the front left wheel for the case of vehicle traveling along the straight asphalt road at 50 km/h speed is shown in Fig. 4. The appearance of measurement signal in Fig. 4, point to stochastic excitation of vehicle oscillation due to road roughness. Obtained measurement records were used as a starting basis for further data processing at different levels and in different domains depending from structure of used identification model.



a) b)

Fig. 3 *Vertical acceleration sensor at the centre of, a) rear left wheel, b) front left wheel.*

In this sense, in Fig. 5 is presented excitation spectra from road roughness identified on the basis measurement signal in Fig. 4. Fig. 6 shows input–output signal in transfer channel: vertical acceleration of front left wheel and suspension connection point with vehicle body, as well as magnitude and phase lage of this channel, marked as AFK, FFK, respectively. These characteristics contain partial information in frequency domain on damping and filtering properties of the suspension system of front left wheel, then resonant frequiencies, factor gain, phase shift, effects of mechanical and functional couplings of the observed sub-system, superposition of low frequency oscillation of sprung mass and high frequency oscillation of unsprung mass.

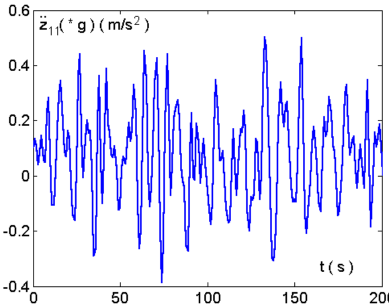


Fig. 4 *Measurement record of vertical acceleration at the centre of the front left wheel.*

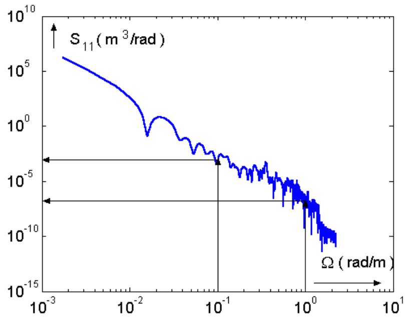


Fig.5 *Identified vehicle excitation spectra due to road roughness on the front left wheel*

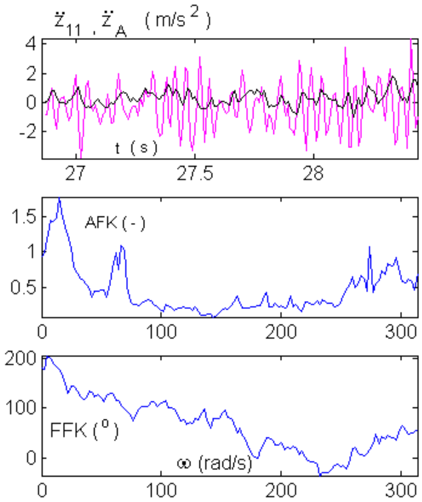
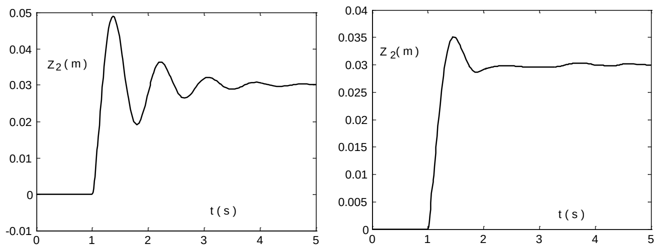
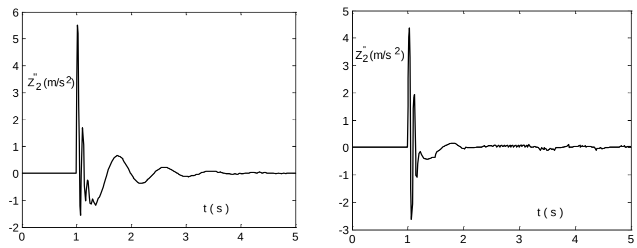


Fig. 6 *Partial transfer function of acceleration between the centre of the front left wheel and suspension connection point with vehicle sprung mass*



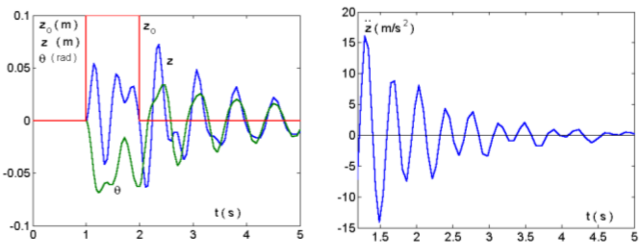
a) b)

Fig. 7 *Displacement of vehicle body, a) linear damping, b) non-linear damping (quarter model)*



a) b)

Fig. 8 *Vehicle body vertical acceleration, a) linear damping, b) non-linear damping (quarter model)*



a) b)

Fig. 9. *a) vertical, z and angular displacement, θ of vehicle sprung mass, b) vertical acceleration of vehicle sprung mass, (impulse excitation, model "2D")*

Characteristic simulation results for quarter model of front left wheel are present in Fig. 7 and 8. For this purpose has been used so – called qurter model, which structure is derived from structure of the vehicle spatial model , in Fig. 1., and differential equations from mathematical models (1) to (6). Random excitation from road roughness was simulated with a filtered white noise source according to obtained excitation spectra in Fig. 5, whose power – spectral density curve when plotted on logarithmitic co – ordinates may be approximated by a straight line with the exponential equation :

$$S(\Omega) = S(\Omega_0)\Omega^{-W} \tag{7}$$

where are :

$S(\Omega_0)$, Ω_0 , W - parameters of road roughness,

Ω - spatial frequency

By means given random excitation signal and registered output signals the components of transfer function , gain and phase angle were determined in frequency domain. The performance parameters representing comfort, suspension working space and tyre load fluctuations are then calculated by means of the input spectral – density and relevant frequincy response function in the frequencies range from 0.25 to 15 Hz. From these relations it can be concluted that:

1. the main levels of body acceleration, as measure vehicle comfort, are concerned in region of body resonance where is the spring the main force transmiter ,
2. the extreme values of body relative displacement, as measure of used workspace also occurs at body resonance,
3. the main shock absorber force and tyre load force fluctuation occure at higher freqencies.

From above analysis follow conclusion, that it would be advantageous to incorporate the shock absorber with greater damping at low frequencies. In this sense, a procedure to selection the optimum form of shock absorber characteristics with respect to vehicle comfort is proposed. An inverse algorithm was used , with input variables, road roughness displacement and weighted body acceleration, and with output variables, displacement of unspring mass and shock absorber force. As result of inverse simulation was obtained the shape of an averaging damping characteristics as unsymmetrical, nonlinear, with optimum ratio of damping rates on the rebound and bump strokes. And finally, as results application one such shock absorber are improved vehicle performance.

The curves in Fig. 7a and 7b, shown vertical displacement of sprung mass, ie., vehicle body, for linear and nonlinear shock absorber characteristics , respectively. Corresponding curves of vertical acceleration are presented in Fig. 8a and 8b.

In relation with improvement of vehicle oscillatory performance are simulation results presented in Fig. 9, for two–dimensional vehicle model.

Fig. 9a provides a comparative overview of time history from which it is obvious, that during the interval of impulsive excitation of the front wheels, the observed transient processes are phase-asynchronized. In doing so, the vertical displacement of center of gravity, z, is changing it's sign, while the pitching angle θ, for the time it retains a negative sign. After the excitation is over, processes are oscillatory damped and phase-synchronized. According to presentaion in Fig. 9b, vertical acceleration (as well as angular, not shown in fig.) of vehicle sprung mass, during initial phase of transient process can achieve high values.

4. CONCLUSIONS

Most of the published results of investigation into the vehicle handling characteristics concern to simplified models, subjected to pure harmonic or transient excitation. But, in reality, the vehicle as complex non-linear dynamical system subjected to random excitation from the environment. Roughness of the road surface is the most important source excitation of the vehicle, and is a major cause of the discomfort of vehicle, fluctuation in the wheel loads, dynamical load of the elements and components of vehicles, but also lead to fatigue of the driver and passengers. In this sense, the modeling and identification of the vehicle oscillatory processes contribute to solution above mentioned problems. The results obtained in this paper shown, that, vehicle response to excitation from the road roughness can be modelled by developed vehicle linear and non-linear mathematical models and previous identified basic road statistical characteristics in form auto-power density functions, as power spectra.

Identified frequency response characteristics contain information on damping and filtering properties of the suspension system, then resonant frequiencies, factor gain, phase shift, effects of mechanical and functional couplings etc., as important indicators of vehicle properties with respect to oscillatory processes.

REFERENCES

[1] H.Appel, and T.Meissner, "Grundlagender Kraftfahrzeug technik II". TU, Berlin, 1998.

[2] M. Mitschke, "Dynamik der Kraftfahrzeuge".1972.

[3] J. Wong, "Theory of ground vehicle". Third edition, John Wiley&Sons, 2001.

[4] J.T.Broch, "Mechanical vibration and shock measurement". The application of the Bruel & Kjaer Measuring Systems, 1976.

[5] R. Radonjić, "Identifikacija dinamičkih karakteristika motornih vozila ", Mašinski fakultet, Kragujevac, 1995

[6] A. Janković, "Dinamika auitomobila", Mašinski fakultet, Kragujevac, 2008.

[7] D. Miloradović, "Istraživanje mehaničkih i funkcionalnih sprega dinamičkog sistema vozila sa aspekta zahtevanih performansi". Doktorska disertacija, Fakultet inžinjerskih nauka, Univerzitet Kragujevac, 2012.

[8] N. Mansfield, "Human response to vibration". 2005.

[9] D. Simić, "Grundlage des Schwingungskomfort". Dissertation TU Berlin D83, 1970.

[10] R. Radonjić, B. Aleksandrović, D. Radonjić, A. Janković, M. Praščević, "Influence the characteristics of the vehicle on its ride comfort". 24th International Conference Noise and Vibration, Niš 2014.

[11] R. Radonjić, D. Miloradović, D. Radonjić, "An approach to vehicle research". International Journal "Mobility & Vehicle Mechanics", Vol. 41, No 4, 2015, pp.53 – 64.

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VIBRODIAGNOSTICS AUTOMATED SYSTEMS

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Abstract - *Automation vibrodiagnostics system is an indispensable and essential segment of modern diagnostics, a large number of information and the processing speed of the parameters is of paramount significant for the adoption of real and reliable diagnosis of the state of the technical system. The development of microprocessor technology, and based on her digital signal processing, has enabled a significant step forward in the protection of machinery with very financially acceptable investment. Direct monitoring and analysis of the situation is obtained good insight into the state machine. Direct monitoring and analysis the situation is obtained good insight into the state of the machine and carry out maintenance when is really necessary implemented. In this way the machines and the process in general resulting an increase in the availability and utilization of machines and plants, reducing costs and increasing profitability as a condition for market operations.*

Keywords: *automation, computers, diagnostics, vibration monitoring.*

1. INTRODUCTION

Under present business conditions, the competitiveness of industrial enterprises in the world market is achieved favorable ratio of price and product quality, and delivering products on time, which is directly related to the state of the production equipment and maintenance of technical systems. Instead of the traditional maintenance based on the a fixed interval maintenance activities, is increasingly being applied to the maintenance of the state, based on diagnostic testing of components of technical systems, which enable the detection of defects at an early stage, thus creating conditions that, engaged in maintenance at the right time, and thereby preventing failures and accidents, and reduce costs.

In this way ensures operation of technical systems without or with as little downtime as congestion, especially unplanned, can cause high costs, may lead to additional damage to other parts of the system which may endanger the safety of people.

Such use of diagnostic tests can help to achieve positive efkti in terms of: increasing the reliability of the technical systems in the process of exploitation, increasing safety, then reduce maintenance costs (through reduction of maintenance activities and optimal planning of maintenance activities, etc.), and thus reduction of total costs, which leads to increased productivity and efficiency.

Today, the Technical Systems (machinery and equipment) industry are becoming increasingly complex, more productive and more expensive, and they are required high level of reliability and minimizing the situation in cancellation, up to nearly complete elimination of cancellation. Due to the cancellation of only one part of the technical system, for example. critical equipment may come to a halt the entire production line with more machines, highly productive machines and the like. (If there aren`t parallel technical systems, which provide greater reliability of the process, but significantly increased financial investment) and, consequently, high costs, environmental damage and endangering the safety of workers, which is especially noticeable in the process industry (petrochemicals, refineries, pulp production, cement or sugar, smelters, etc.).

For these reasons shouldn`t be allowed to stop the damage and critical machines, which also means stop and stop the entire production (eg. Turbine in the heating plant), or whose disruption can cause outages entire drive.

2. COMPUTERS AND AUTOMATION

Need immediate more continuous monitoring diagnostic parameters of technical systems in order to detect their possible defects at an early stage and make it create the conditions to ensure timely undertaking of appropriate maintenance activities prevent failures and disasters, is contributed to the introduction of automation and computer.

If in such circumstances the collection and processing of data, as well as diagnostic decision-making exercised classic (no automated) procedures, diagnostic tests could become very complex and time-consuming , because a person in such situations usually wouldn`t be able to have speed and accuracy.

In order to avoid this is done the automation of diagnostic processes, or instead of traditional diagnostic methods used automated diagnostic systems based on the use of computers and other modern information and communication technologies. In such cases, all of these operations are performed by a computer, and the computer controls the entire diagnostic process, Figure 1, which greatly simplifies the process and speeds, so that the reports (protocols) on conducted diagnostic tests can be obtained immediately after conducted test.

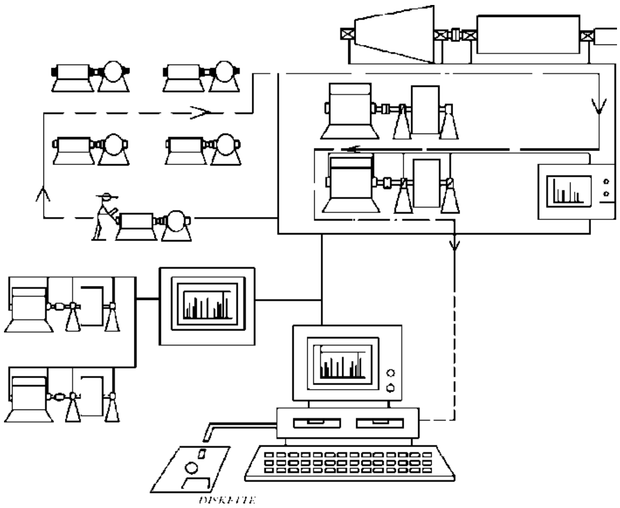


Figure 1. *Configuration of an automated diagnostic system for diagnosing the condition of a manufacturing plant [1]*

Automation is by definition the use of "automata" for the implementation of a process. The word automat Greek origin and means a device which by itself performs a process. Man this isn`t totally excluded from the process, but it is role is reduced to a minimum, ie. only to initiate, control and stop processes. Automation therefore greatly simplifies a process and frees people from excessive involvement in this process.

With the development of computers, sensors and frequency analyzers furthers the application of expert systems in technical diagnostics. Most developed PdM system today is based on monitoring changes in the frequency spectrum.

When PdM experts estimate the frequency spectrum, they do so by observing a process of evaluation linking the same basic symptoms observed in the spectrum of the mechanical elements of the machine. These identified symptoms or characteristics usually include the general level of vibration, changes with respect to the reference measurements, including noise level in the spectrum, harmonics, etc.

In order to bring it closer to those skilled, expert system uses the same features to evaluate the spectrum and to perform some basic conclusions. It is obvious that an expert system can`t perform observations in the same manner as the expert, but instead is calculated using a limited frequency symptom database, the analyzer and the nominal value of the input RPM user.

However, in practice there is the problem of harmonizing the differences between the theoretical calculations of symptoms related to the machine parameters such as RPM and the actual existence and location of these symptoms in the spectrum.

In “hand” inference, this can be solved by using the method of compensation speed as as the search (commissioned analysis) or cross-linking. These techniques make it look like the spectrum of visual evaluation looks fixed. However, when “automatically” copies some conclusions, there is a different situation.

As previously mentioned, the symptoms of which are used in expert system PdM base, as a rule, the peaks in the spectrum of the vibration corresponding to the characteristic frequencies of the machine - eg., The amplitude of the working speed (1 ×), the amplitude of the second harmony working speed (2 x), increase energy side group around the blade passing frequency (BLPF). When set turbogenerator, typical parameters for which monitoring should be included: X/Y axis vibration, axial position and temperature lubricating oil for bearings. Although they achieved different levels of success, it can be concluded that the majority of expert systems that are available today can not replace human experts, but it can help in making the diagnosis of machine condition.

In recent years, the development of microprocessor technology, and based on her digital signal processing, enabled significant progress in the protection of machinery with very financially acceptable investment. Direct monitoring and analysis of the situation is obtained good insight into the state of the machine and carry out maintenance when the action is really necessary, that is. implemented in accordance with the state of the machine. In this way we manage in increasing the safety and efficiency of machines and plants, reducing costs and increasing profitability, which is a prerequisite for market operations.

A large number of mechanical systems during operation is dynamically only a small number is capable of for static stresses. It therefore of particular interest to exert control over dynamic systems whose defects can cause damage and threaten the lives of workers.

The development of microprocessors and digital signal processing allows for the tracking and display of all relevant data in the appropriate format (format), speaking about the real state (health) of mechanical systems.

Thus developed control system allows you to manage mechanical systems and process as a whole, resulting in increasing the safety of machinery and plants, in general, reduces costs and increases profitability, which is a prerequisite for market operations.

Due to data availability, system for collecting data, this system allows the display of data in the relevant form (format) from which a decision is made on the condition of the machine and take appropriate action to maintain. With the existence of a system for the collection (and analysis) of data must be relevant and organizational data management system on the basis of which decisions are made. Direct monitoring and analysis of the situation gets an insight into the state of the machine and carry out maintenance actions. In this way we manage the machine as a whole, resulting in increasing the availability and utilization of machines and plants, reduces costs and increases profitability, which is a condition for market operations.

3. MONITORING EQUIPMENT

Parameters monitoring systems are indirect individual, associated with structural parameters (vibration, temperature, backlash in the bearing, oil pressure, etc.), And the holders of the accurate information about the technical condition of the system. Parameters that carry the most information about the state of the technical system are certainly vibration parameters, among them also be the parameters of displacement, temperature, noise, power parameters, the parameters of the lubricating oil and others.

A large number of techniques is available from the monitoring of rotating system which will to be chosen depends on the depth needed to diagnose, with economically justifiable investment [8].

Let us mention some of the most commonly used technique for monitoring these types:

1. **vibro-acoustic analysis:**
   - vibrodiagnostics analysis: analysis of the overall level of vibration, spectral analysis, phase analysis, vector real-time analysis, orbit, DC analysis, trending parameters, the shock pulse method (SPM method), Energy analysis, Zoom FFT analysis, CPB analysis, cepstral analysis, SED detection, detection of HFD, LFD detection, SEE technology,
   - analysis of noise.
2. **analysis of operating parameters:**
   - energy analysis,
   - the thermal analysis,
   - analysis of transfer and spread in the system,
   - analysis of technological parameters (flow, pressure, technological parameters of water, etc.)
3. **wear debris analysis and combustion:**
   - analysis of oils and lubricants (atomic spectroscopy, infrared spectrophotometry, membrane filtration, gas chromatography, etc.).
   - gas analysis.
4. **monitoring corrosion:**
   - visual methods (Visual microscopy, metallography, etc.)
   - gravimetric method (pH-metry, AAS method, volumetry, etc.)
   - electrochemical methods.

In recent years, most modern portable measurement devices like the digital (computer) making a connection to the sensor is achieved by using the A/D convector.

The basis of a control - measuring chain consists of:

- encoders (sensors) vibration,
- measuring and analytical units and
- vibro-diagnostics computer system for monitoring and analysis.

CONCLUSION

Progress computer and measurement techniques, and continuous reduction in price development of computer hardware, allows today a significant step forward in the protection of mechanical systems with very financially acceptable investment. To achieve this goal will be equipped with generators in order to timely detect errors in operation, following changes in the state machine, then set depending on the operating modes, diagnose the causes of failure and preserved after the events, important information about the condition. Such devices for monitoring the status confirmed in large numbers around the world and will remain, an essential equipment also in the future, for both new and old facilities.

REFERENCES

[1] Adamović, Ž.: *Maintenance technology*, Technical Faculty "Mihajlo Pupin", Zrenjanin, 1998.

[2] Adamović, Ž., Ilić, B.: *The science of technical system maintenance, Serbian Academic Center,* Novi Sad, 2013.

[3] Adamović, Ž., Ašonja, A.: Milošević, D., *Vibro-tribology-diagnostics and diagnostics of machines*, Serbian Academic Center, Novi Sad, 2012.

[4] Ilić, B., Adamović Ž., Savić, B., Stanković, N.. *Development of a model of an automated diagnostic system and its impact on the reliability of technical systems*, Technical diagnostics, Banja Luka, vol. 4, br. 1-2, pp 20-40, 2012.

[5] Knuth, T.: *Early detection of damage using vibration analysis, maintenance of machinery and equipment* - OMO, Beograd, 1988.

[6] D. Spasić, Ž. Adamović, S. Meza, G. Otić, *Automation of technical diagnostics, expert systems in diagnostics and robots maintained,* XV Conference "Technical diagnostics of machines and plants « Serbian Technical dijagnostic society, work No.18, Vrnjačka Banja, 2015.

[7] Ž. Adamović, D. Spasić**,** P. Alargić, S. Meza, N. Nikolić, G. Otić, *Maintenance-Methodology-Reliability-Diagnostics-technologist.*, Society for Technical diagnostics Serbia, Smederevo, 2015,

[8] Adamović, Ž., Đapić, M.: *Engineering Mechanics*, Part 3, Dynamics, Faculty of Engineering "Mihajlo Pupin", Zrenjanin, 2004.

[9] Ž.Adamović, Lj. Josimović, S. Vulović, B. Ilić, D. Spasić, *Vibration diagnostic and maintenance of technical systems,* The Society for Technical Diagnostics Adam institute, Smederevo, 2016

[10] D. Spasić, Ž. Adamović, N. Jeftić, N. Stefanović, S. Cvetković, *Non-destructive methods of diagnosis*, Publishers Weldspad end Newpress, Smederevo, 2013.

[11] D. Spasić, N. Jeftić, Z. Janjić, Ž. Adamović, Electrohydraulic system for automatic gage control (agc) for tandem cold mill plant in steelworks Smederevo, Journal of the Balkan Tribological Association, Book 1. Vol. 22. 2016.

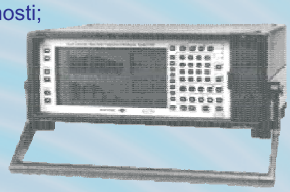
[12] D. Spasić, D. Malić, S. Meza, N. Jeftić, *Total Productive Maintenance in our industry*, XI Conference "Technical diagnostics of thermal power plants, solar power plants, heating plants and hydro power plants", Proceedings, Work 004, Vrnjačka Banja, 2014.

CENTAR ZA TEHNIČKU DIJAGNOSTIKU & LABORATORIJA ZA BUKU I VIBRACIJE

UNIVERZITET U NIŠU
FAKULTET ZAŠTITE NA RADU U NIŠU
Čarnojevića 10a, 18000 Niš
Tel.: 018 529-747;   Fax: 018 529-748

Merenje i analiza nivoa buke

- Određivanje ekvivalentnog i merodavnog nivoa buke u zavisnosti od karaktera buke i vremena izloženosti;
- Frekvencijska analiza nivoa buke;
- Statistička analiza nivoa buke;



Merenje i analiza nivoa vibracija

- Određivanje merodavnog nivoa vibracija;
- Frekvencijska analiza nivoa vibracija;

Merenje i analiza akustičke aktivnosti mašina

- Određivanje nivoa zvučne snage u realnom ambijentu merenjem intenziteta zvuka;
- Određivanje nivoa zvučnog pritiska merenjem intenziteta zvuka;

Preventivno održavanje mašinske opreme

- Praćenje stanja mašinske opreme na osnovu nivoa zvuka;
- Praćenje stanja mašinske opreme na osnovu nivoa vibracija;
- Balansiranje rotirajućih elemenata mašinskih sistema;

Projektovanje vibroakustičke zaštite

- Projektovanje sistema za izolaciju i apsorpciju zvuka;
- Projektovanje sistema za vibroizolaciju;
- Projektovanje akustike prostorija;

Merenje i analiza akustičkih karakteristika prostorije

- Vreme reverberacije;
- Izolaciona moć pregradnih zidova i konstrukcija:
- Izoalaciona moć od zvuka udara;
- Koeficijent apsorpcije materijala;

Sistematsko merenje komunalne buke

- Frekvencijska, statistička i procentualna analiza nivoa buke u komunalnoj sredini;
- Zoniranje urbanih prostora u odnosu na nivo buke;
- Izrada strateških karata buke;
- Ispitivanje uticaja akustičke aktivnosti instalirane opreme u javnim objektima na životnu sredinu;

Obrazovanje

- Instruktivni seminari;
- Studije za inovaciju znanja;
- Specijalističke studije;

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VIBRATIONS AS PARAMETER SHEET OF MACHINE

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Abstract - *Using the frequency analysis, we are able to find out source of unwanted vibration on almost all machines. The the diagnostic approach and starts from the fact that each cause of the disturbing vibration generates clearly defined characteristics, recognizable primarily by beam as a key parameter. Single, wideband vibration measurements are a useful indicator of vibration, which can be used for example when assessing the general condition of the machine.*

Keywords: *vibration machine, measurement, diagnostics.*

INTRODUCTION

Vibrations which occur on machines represent the movement of mechanical components machines forward back and forth as and their reaction to the internal and external of force, or oscillating a rigid body relative to the their equilibrium position. Causative agent disturbance of movement is a force that can be determined, random vibration we call a the free periodical movements of the body which is performed after termination of the operation of disturbance force.

GENERAL FEATURES

The lately developed is a completely new technology of measuring vibration and sound (of noise), which because of its generated content is one of the most important of indicators the overall indicative of the dynamic state of the machine and systems and system components. With the help of measurement of the vibrations may monitor the status of a large number of machines.

Under the to mechanical vibration, imply a oscillating motion of a rigid body relative to the his equilibrium position.

The causative agent the force that forced by its nature can be determined by or accidental. A free vibration is called periodic motion which the body performs to cessation of force.

Vibrations can be displayed in two domains: time (π1) and frequency (π2). Between both areas there is no absolute correlation. The practical reasons prefer to form in the frequency plane (π2), shown in Figure 1.

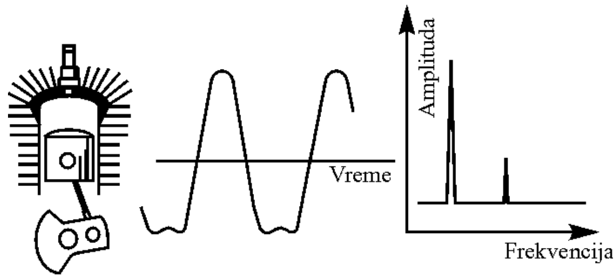

Figure 1. *Comparative presentation of characteristic signals in the time and frequency domain in the case of a piston mechanism*

CAUSES OF VIBRATIONS

With regard to the character and nature of creation, mechanical vibrations are generally divided into two groups: coercive, its own.

Forced vibrations are caused by the action of dynamic forces, the system changing its direction or size.

Each source generates vibrations. The list of potential causes of disturbance vibration can be:

- unbalance mass of rotating parts of the system,
- insufficient dynamic stiffness of the chassis and the foundations of the system,
- disorder centricity joints and bearings,
- deflection shaft (mechanical indulgence),
- shabby, eccentric or damaged gears,
- bad driving belts and chains,
- faulty roller and plain bearings,
- electromagnetic forces
- aerodynamic forces,
- hydraulic forces,
- uneven clearance between the rotor and stator motor,
- looseness in the joints,
- pass through resonance,
- scuffing, etc.

The vibration must be a force that is changing or at its direction or according size. Any cause of vibration has its own characteristics.

Own vibrations are a function of the physical constants of the mechanical system (mass, stiffness, damping). forced vibrations cease removing the disturbing force, but on its own vibrations it is possible to operate only through structural changes system parameter.

FREQUENCY ANALYSIS

Body vibrates when describing oscillatory movement around the reference position. Number of full-cycle movements in a period of one second is called frequency and is measured in hertz [Hz]. In practice, the vibration signals usually consist of very many frequencies that are happening simultaneously, so that we can no immediately see the wrong amplitude changes in time, as far as components are present and what their frequency. The interpretation of the vibration signal individual components is called frequency analysis, a technique be can regarded a the basis in the diagnosis of the vibration.

When we perform frequency analysis of vibrations in machines, normally we find a large number of prominent periodic frequency components that are directly related to the basics of navigating the various machine parts.

QUANTIFYING THE LEVEL OF VIBRATION

Amplitude vibration describes the strength of vibration and can be quantified in several ways.

The value from peak to peak is useful in that it shows the maximum distance between the waves, which is useful when the size of, for example, move vibration of mechanical work an important for consideration maximum load or mechanic reaction.

The effective value of the amplitude is the most appropriate because it takes into account the history of wave in time and gives a value of amplitude that is directly related to the energy content, and hence the destructive possibilities of vibration.

USE OF MEASURING VIBRATIONS

Individually, wideband vibration measurements vibration are a useful indicator of vibration, which is used to assess the general state machine. The measured actual level will be assessed by comparison with a prior or later or measured levels of published criteria. Impractical is to change frequency (speed shaft, the ratio of the number of teeth) by using other methods to reduce unwanted vibration levels. For example, the re-adjustment of machine parts (by changing its resonant frequency) by changing its mass or stiffness, weakening the transfer of vibrations by using insulating materials in order to reduce the amplitude of vibration.

LEVEL OF COMPLEX VIBRATION

Vibration component parts of the system are complex and consist of multiple frequencies. Mostly a total shift shall be the sum of all individual vibration. Where there is vibration complex, representation apply the to a diagram of vibration level, should first determine the individual displacements and their frequency. This is done with the help of vibration analyzers with adjustable filter.

The signal is recorded as characteristic vibration of the measuring point and it is the summation of a large number of individual sources and generally contains in itself a lot of complex content.

Measurements of the vibration acceleration are closely linked with the inertial forces acting on the system in which they can occur relatively large forces at high frequencies, although the vibration displacement and velocity may be small. Excessive force may result in termination of lubrication and can damage the bearing surfaces. Basically, the measurement of vibration acceleration is recommended for vibration frequency above 60,000 cycles per minute, although they can be used and measurement speed.

In addition, when assessing the level of vibration to be measured and the stage, which allows the application of most favorable ways for compare of one with the other movements. Comparison of the relative motion of two or more parts of the system, it is often important in the diagnosis of specific defects on components of the system. For example, if analysis reveals that vibration of a part of the system are out of phase with the the fundamentals, ie their absolute vibration must be checked cause of looseness of screws, watering concrete and others.

The measurement phase is important for balancing (balancing weight). If the problem is part of a system imbalance, whereby they can measure the phase, then we can in this case to balance the work.

The goal is not to determine what level of vibration a part of the system can withstand so that they can be eliminated before the failure. Absolute tolerance limits of vibration or for any part of the system are no possible. Analysis of malfunction and cancellations is quite complex to make such boundaries could exist. Experience specialist vibration may help to obtain some realistic guidelines.

In determining the acceptable levels of vibration , to consider the experience and factors such as safety, cost of removing defects, costs due to the halt in production, the importance of the system in the technological chain of production and others.

VIBRATION MEASUREMENT

In practice today for vibration measurement commonly used two types of instruments, such as:

- magnetic electric PIEZO instruments, instruments which are connected with magnetic base on the bearing housing or another convenient location and
- instruments based on the principle of the probe (is the hand during operation) long about 100 (mm), the measurement is performed on the measuring point.

The vibrations are recorded by an analog-converting mechanical displacements into analog electrical signals (current or voltage). Instrument, designed for operation under operating conditions, is made up of converters, measuring instrument with an amplifier and filter. The configuration allows for measurement of all three vibration characteristic values. Example of a diagnostic instrument called PCE-VT204, is shown in Figure 2. The device therefore can be used to test the vibration engines, transmissions, gears, wheels, etc. But also for measuring rotation. Associated acceleration sensor can be placed directly or magnet in combination with the screw supplied for vibrometer.

Figure 2. *Vibrometer PCE-VT204*

Converters vibration can be contact and non-contact, wherein is first measuring absolute and another relative vibration. In addition, their selection also affects the ratio of the original signal strength and conductivity connections to the measuring station. Transducer can measure the acceleration of vibration (piezoelectric), speeds (inductive) and movements (seismic).

Contactless transducer are usually performed on an inductive or capacitive principle. Measurement of vibration and noise and processing of the measurement results are only the first part of the second block in a logical sequence: detection, analysis, intervention.

The second part refers to identification of the dominant sources of registered in the area of of time or of frequency , using a synthesis of all knowledge about the specific characteristics of the potential oscillation of, most commonly rotating mechanical parts.

In all stages of the proceedings frequency identification is a key parameter.

Instruments for vibration measurement can be understood as:

- measuring instruments (for periodic checks of vibration),
- control instruments (for continuous control of vibration),
- analyzers (adjustable filter to extract individual frequencies from complex vibrations).

ANALYSIS VIBRATIONS

Signal the vibrations recorded on the surface of the machine part (assembly) are present traces of many individual signals generated in places where there is a transformation of the work of useful energy into the energy of vibration. For the purpose of identifying such sources is not enough only a comparison of overall vibration levels, already should be carried out carefully analysis on the component parts and examine the each a part of.

The process of separating the frequency is called frequency analysis, as its end result is the aforementioned frequency spectrum. This is achieved by filtering the signal vibrations in the instrument , which is called analyzer.

More complete analysis is always performed in laboratory conditions, using high-quality analysts who have extensive features amplitude modification and frequency transformation. Signal processing is usually performed by analog. Lately, it is more concrete and digital operation. Snadbeven fast processors, these analyzers offer a wide variety of functions in the analysis of dynamic phenomena in all three of domains were used: amplitude, time and frequency.

Vibration Analysis is performed when periodic monitoring of overall vibration or noise on mechanical systems revealing substantial increase. This analysis should also be executed on commencement of maintenance program according to the state, to determine the technical condition of the system. Procedure of analysis can be divided into two phases:

- Data collection and
- Interpretation and processing of data (comparing the recorded data with basic data on the components and / or systems).

ACCESS CONTROL AND TECHNICAL DIAGNOSTICS BY VIBRATION

The overall diagnostic approach starts from the fact that each causative agent the disturbing vibration generates with predefined character, recognized primarily by beam as a crucial parameter.

In so doing, the partial identification of the dominant component, registered areas or frkventnog time domain, it is placed on the use of assimilated status and experience of the specific characteristics of the moving image oscillation movable, usually rotating machine parts. For a multitude of vital rotating, mechanical parts, despite the fundamental frequency and its harmonics, exact mathematically defined by a series of forced and natural frequencies that may suit the dominant component in the recorded spectra.

The mere physical interpretation of the characteristic frequency can be simulated via recorded extraordinary orbital trajectory of the rotor. The shape and orientation of the trajectory of of the rotor in a recess formed summary vector of external and internal forces.

The causes that influence the occurrence of vibrations can be mentioned: bad design solution, incorrect installation of the entire assembly, inadequate maintenance due to improper dijagosticiranja, abnormal (excessive) the work of all parts of the rotating assemblies, poor fixation and misalignment basis for the same, necentriranje joints, poor lubrication of bearings, the appearance of various contaminant in the lubricant, Improper installation of bearings, inner ring of the bearing from the shaft turns, the outer ring turns with housing, uncontrolled friction between the shaft and the other machine parts, neučvršćenosti individual machine parts, unbalanced rotors, cavitation on pumps, gear tooth damage, pressure and shock loads that occur during operation, misaligned shafts, etc.

CONCLUSION

The effect of vibration on the machines

On parts of machines and other equipment vibration impact is manifested in various ways, such as:

- the occurrence of plastic and elastic deformations, fractures and accidents due to fatigue especially in the areas of stress concentration,
- increased friction,
- the greater an energy loss and lower the working effects machines.

From the aspect of the functionality of machines and other electronic equipment on machines, vibrations are harmful.

According to the level of amplitude and frequency of the equipment must comply with the limit values. Otherwise, there are: functional errors, improper operation, a bad product (intermediate product) and the like. As the cause of all of this in certain systems may arise following problems:

- avoiding to excessive turbine vibration bearings leads to increased workload and shorter working life,
- level of vibration above the prescribed limits adversely affect the operation of electric motors and generators,
- the quality of piece grinding machines can only be achieved when the vibration within the prescribed limits and
- the change in vibration is particularly sensitive electronic equipment (exceeding the allowed level of vibration leads to the appearance of physical contact and changes characteristic of of electrical circuits).

In these cases, when the measured level of vibration is exceeded, must to undertake certain measures to reduce or eliminate vibration. Permissible level of vibration is determined by rules for the respective machines.

REFERENCES

[1] Adamović, Ž.: *Maintenance technology*, Technical Faculty "Mihajlo Pupin", Zrenjanin, 1998

[2] Adamović, Ž., Ilić, B.: *The science of technical system maintenance, Serbian Academic Center,* Novi Sad, 2013

[3] Adamović, Ž., Ašonja, A.: Milošević, D., *Vibro-tribology-diagnostics and diagnostics of machines*, Serbian Academic Center, Novi Sad, 2012

[4] Ilić, B., Adamović Ž., Savić, B., Stanković, N., *Development of a model of an automated diagnostic system and its impact on the reliability of technical systems*, Technical diagnostics, Banja Luka, vol. 4, br. 1-2, str. 20-40, 2012

[5] Knuth, T.: *Early detection of damage using vibration analysis, maintenance of machinery and equipment* - OMO, Beograd, 1988

[6] D. Spasić, Ž. Adamović, S. Meza, G. Otić, *Automation of technical diagnostics, expert systems in diagnostics and robots maintained*, XV Conference "Technical diagnostics of machines and plants « Serbian Technical dijagnostic society, rad br.18, Vrnjačka Banja, 2015

[7] Ž. Adamović, D. Spasić**,** P. Alargić, S. Meza, N. Nikolić, G. Otić, *Maintenance-Methodology-Reliability-Diagnostics-technologist.*, Society for Technical diagnostics Serbia, Smederevo, 2015

[8] Adamović, Ž., Đapić, M.: *Engineering Mechanics*, Part 3, Dynamics, Faculty of Engineering "Mihajlo Pupin", Zrenjanin, 2004

[9] Ž.Adamović, Lj. Josimović, S. Vulović, B. Ilić, D. Spasić, *Vibration diagnostic and maintenance of technical systems,* The Society for Technical Diagnostics Adam institute, Smederevo, 2016

[10] D. Spasić, Ž. Adamović, N. Jeftić, N. Stefanović, S. Cvetković, *Non-destructive methods of diagnosis*, Publishers Weldspad end Newpress, Smederevo, 2013

[11] D. Spasić, D. Malić, S. Meza, N. Jeftić, *Total Productive Maintenance in our industry*, XI Conference "Technical diagnostics of thermal power plants, solar power plants, heating plants and hydro power plants", Proceedings, Work 004, Vrnjačka Banja, 2014

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ON THE DYNAMIC BEHAVIOUR OF A FOLDED PENDULUM MECHANISM

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Abstract - *A folded pendulum mechanism, which has been proposed for vibration isolation purposes, is considered in this study. It consists of two mathematical pendula: the first one is a classical pendulum, which is stable, and the other one is an inverted pendulum, which is unstable. These two pendula are mutually connected by a weightless rigid slender bar. Given the fact that certain geometric constraints exist, the analysis of the dynamic behaviour of this mechanism is challenging, both analytically and numerically. The exact form of the potential energy is derived and investigated in detail. Lagrange's equations of the first kind with undetermined multipliers are used to form the equations of motion. Numerical analyses are carried out to provide an insight into possible dynamic behaviour of this mechanism, distinguishing between the desirable and undesirable one. Some discussion of the case when the oscillations are small is also given.*

1. INTRODUCTION

This study is concerned with the so-called folded pendulum mechanism (FPM) (Fig. 1a). This mechanism was proposed in 1993 by Blair et al. [1]. This type of mechanism consists of two pendula, one of which is a classical pendulum (Fig. 1b), i.e., its gravity force always tends to pull it back to the vertical stable position; the other one is an inverted pendulum (Fig. 1c), whose gravity force tends to move the pendulum away from its equilibrium. These two pendula are mutually connected pivotally with the third element, which may be a slender bar, platform or any other suitable linkage (Fig. 1a). As a consequence of the permanent mutual counterbalancing of the gravity forces of the two pendula, this mechanism can perform oscillatory motion with a very low natural frequency.

This possibility of achieving a low natural frequency makes this mechanism very suitable for a variety of applications. In [1], the mechanism was proposed to provide ultra-low frequency vibration isolation for a laser interferometer gravitational wave detector. In [2], the theoretical and experimental transfer function for the FPM was presented and investigated. It was concluded that the FPM has a similar transfer function as a simple pendulum with a long length, i.e. with a long period. In [3], the FPM was used as a long-period (1-min) seismometer for studying the dynamics of ocean's shore waves.

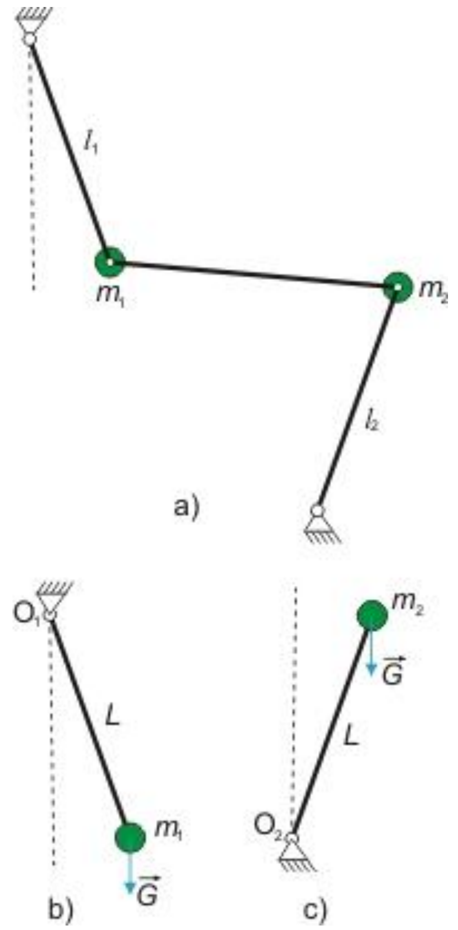

Fig. 1 *a) Folded pendulum mechanism; b) Classical pendulum; c) Inverted pendulum*

It was shown in [4] that the FPM is very sensitive to the low-frequency components of the tilt vibration, and, therefore, it could be used as a tiltmeter. Both the theoretical analysis and the demonstration of the prototype form were given. In [5], a folded pendulum accelerometer was developed to be used as a sensor in multiaxis active seismic isolation systems. Such device is characterised by monolithic design, elimination of the stick-and-slip effect and a very high quality factor.

However, it has been generally observed that the influence of the system parameters on the desired low frequency behaviour is significant. Therefore, better understanding of its dynamic behaviour both in a linear regime and in a nonlinear one, is very important. Thus, this study aims at providing a deeper insight into the dynamic behaviour of this mechanism from both perspectives.

2. EQUATIONS OF MOTION

In this Section, exact equations of motion for the FPM are derived. The model is assumed to contain two discrete masses (particles) m_1 and m_2, which move in a vertical plane (Fig. 2). The length of the classical pendulum is l_1, the length of the inverted one is l_2 and the length of the linking bar is l_3. Other elements of the system are assumed as weightless. To describe the motion of the particles, four coordinates x_1, y_1, x_2, and y_2 are needed, where the Cartesian coordinate system has the origin at the fixed point O.

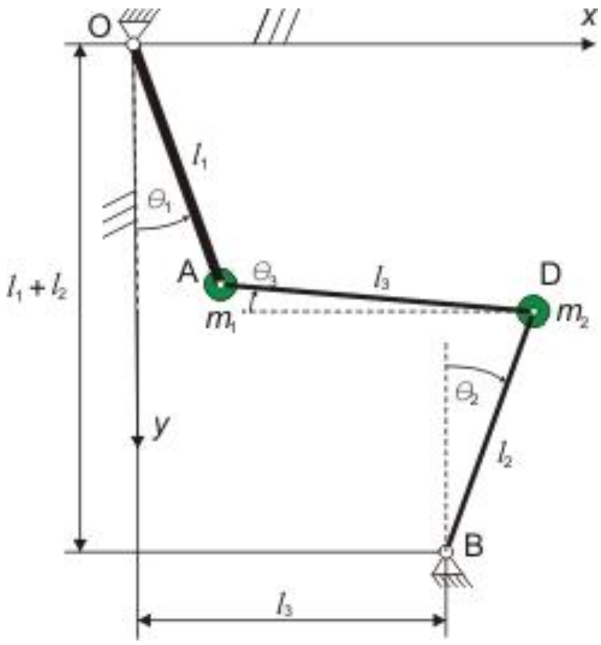

Fig. 2 *FPM under consideration*

The geometry imposes three constraints (see Fig. 2)

$$g_1 = x_1(t)^2 + y_1(t)^2 - l_1^2 \, , \tag{1}$$

$$g_2 = (x_2(t) - x_1(t))^2 + (y_2(t) - y_1(t))^2 - l_3^2 \, , \tag{2}$$

$$g_3 = [x_2(t) - l_3]^2 + [y_2(t) - (l_1 + l_2)]^2 - l_2^2 \, . \tag{3}$$

Thus, the FPM has one degree of freedom. Due to the existence of the constraints, it is suitable to use Lagrange's equations of the first kind. It is known that these equations contain an undetermined Lagrange's multipliers, which will be denoted herein by λ_j. The subscript j corresponds to the number of constraints, so j=1,2,3. A general form of Lagrange's equations of the first kind is

$$\frac{d}{dt}\frac{\partial L}{\partial \dot{q}_i} - \frac{\partial L}{\partial q_i} - \lambda_j \frac{\partial g_j}{\partial q_i} = 0, \tag{4}$$

where q_i includes x_1, y_1, x_2, and y_2; L denotes the Lagrangian $L = T - V$, where T and V are the kinetic and potential energy, respectively.

It should be emphasized that the motion and stability will be presented and investigated with respect to the mass 2, i.e. to the coordinate x_2. This is due to the fact that the inverted pendulum can lose stability of the desired oscillatory motion, so its motion here is of particular importance.

The expressions for the kinetic energy T, the potential energy V and the Lagrangian L are given by

$$\begin{aligned} T &= \frac{1}{2} m_1 v_1^2 + \frac{1}{2} m_2 v_2^2 \\ &= \frac{1}{2} m_1 \left(\dot{x}_1^2 + \dot{y}_1^2\right) + \frac{1}{2} m_2 \left(\dot{x}_2^2 + \dot{y}_2^2\right), \end{aligned} \tag{5}$$

$$V = -m_1 g - m_2 g \, y_2 \, , \tag{6}$$

$$\begin{aligned} L = T - V = \frac{1}{2} m_1 \left(\dot{x}_1^2 + \dot{y}_1^2\right) + \\ + \frac{1}{2} m_2 \left(\dot{x}_2^2 + \dot{y}_2^2\right) + m_1 g \, y_1 + m_2 g \, y_2 . \end{aligned} \tag{7}$$

Using Eqs. (1-3) and Eqs. (5-7), one can write down the Lagrange's equations of the first kind (2) as follows:

$$m_1 \ddot{x}_1 = \left(G^T \cdot \lambda\right)_1 , \tag{8}$$

$$m_1 \ddot{y}_1 = m_1 \, g + \left(G^T \cdot \lambda\right)_2 , \tag{9}$$

$$m_2 \ddot{x}_2 = \left(G^T \cdot \lambda\right)_3 , \tag{10}$$

$$m_2 \ddot{y}_2 = m_2 \, g + \left(G^T \cdot \lambda\right)_4 , \tag{11}$$

where

$$G = \frac{dg}{dp} , \tag{12}$$

and

$$p = \{x_1, y_1, x_2, y_2\} . \tag{13}$$

Now, the following system of exact equations of motion is derived

$$m_1 \ddot{x}_1 - 2\lambda_1 x_1 - 2\lambda_2 (x_2 - x_1) = 0 , \tag{14}$$

$$m_1 \ddot{y}_1 - 2\lambda_1 y_1 + 2\lambda_2 (y_2 - y_1) - m_1 \, g = 0 , \tag{15}$$

$$m_2 \ddot{x}_2 - 2\lambda_2 (x_2 - x_1) - 2\lambda_3 (x_2 - l_3) = 0 , \tag{16}$$

$$\begin{aligned} m_2 \ddot{y}_2 - 2\lambda_2 - (y_2 - y_1) - 2\lambda_3 [y_2 - (l_1 + l_2)] - \\ - m_2 \, g = 0. \end{aligned} \tag{17}$$

Equations (1-3) and (14-17) form a set of three nonlinear algebraic equations and four nonlinear second-order differential equations. Before their analysis, some consideration of the potential energy is carried out in the next Section.

3. ON THE POTENTIAL WELL AND ASSOCIATED BEHAVIOUR

The FPM may have different regimes of oscillatory motion depending on the system parameters. Namely, Fig. 2 implies that there may exist a few positions around which oscillatory motion is performed. Bearing in mind the use of the mechanism in vibro-isolation systems and sensors, of interest here is to investigate the oscillations around the position when the rod OA is vertical (Fig. 2). So, it is necessary to identify those parameters of the system for which the oscillatory motion is stable. To that end, the potential energy of the system given by Eq. (6) is analysed (Fig. 3), by changing m_2 and fixing all other system parameters to: m_1=1, l_1=1, l_2=1, l_3=2 (note that all these values are used subsequently if not explicitly stated differently; the units are omitted for brevity). Depending on the value of m_2, three different shapes of the corresponding potential well are identified. When m_2=0.5 (m_2<m_1), the minimum of the potential energy, which corresponds to the stable equilibrium position, is exactly above the hinge B. The potential energy also has the maximum, which corresponds to the unstable equilibrium placed on the left-hand side with respect to the stable one. When m_2=1.1 (m_2>m_1), both equilibria move to the right-hand side, so that the unstable equilibrium is now placed above B. In Fig. 3, the parts of the potential well that are associated with oscillations around the stable equilibria are shown in green. The shape of the potential well corresponding to the case when the masses are equal (m_2=m_1=1) is related to indifferent equilibria.

The overall potential energy can be investigated by turning back to the fact that it is actually the sum of the potential energy of the mass 1 and mass 2. In other words, the form of the overall potential energy can be viewed as a mutual interaction between these two terms. This fact can significantly facilitate the process of deriving the analytical expression for the potential energy as well as its analyses due to the high nonlinearity of the equations derived. Accordingly, the potential energy of the mass m_1 must have a local minimum, while the potential energy of the mass m_2 must have a local maximum. Thus, the coordinates y_1 and y_2 are first expressed in terms of the *x*-coordinates by using the equations of constraints given by Eqs. (1) and (3) as follows:

$$y_1 = \sqrt{l_1^2 - x_1^2}\ , \tag{18}$$

$$y_2 = l_1 + l_2 - \sqrt{l_2^2 - l_3^2 + 2l_3 x_2 - x_2}\ . \tag{19}$$

Now, by using Eq. (2), one can express x_1 in terms of x_2:

$$x_1 = \Big[-b\,x_2 - \sqrt{b^2 x_2^2 - 4c(-108 - 40a + 184x_2 + {}} \\ {+ 56ax_2 - 127x_2 - 28ax_2^2 + 36x_2^3)} \Big] / (2c)\ , \tag{20}$$

where

$$a = \sqrt{(3 - x_2)(x_2 - 1)}\,,$$
$$b = -46 + 12a + 68x_2 - 32x_2^2\,, \tag{21a-c}$$
$$c = 49 - 56x_2 + 32x_2^2\,.$$

Now, the overall potential energy (6) as well as the one for each mass can be expressed only in terms of x_2. This enables one to present this quantity graphically, as given in Fig. 5. It is seen that V_2 has a local maximum, while V_1 has both a maximum and a minimum. Due to the existence of the maximum and minimum, the potential energy V_1 has a well (the region between the points E and F). It should be noted that this would be potential energy for the whole system for $m_2 = 0$. The influence of the mass m_2 is such that it decreases the width and the depth of the potential well of the system, which is presented by the region between the points H and J. Given the fact that the system is conservative and that the energy conservation law holds, this can be used to determine which initial conditions lead to the desirable oscillatory motion.

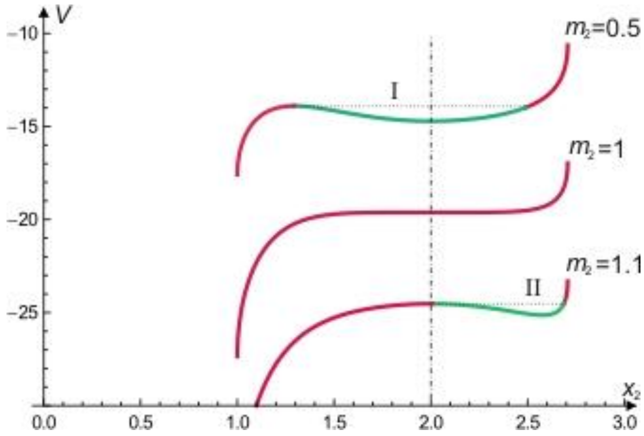

Fig. 3 *Graph of the overall potential energy* V *for* $m_1 = 1$ *and three different values of* m_2

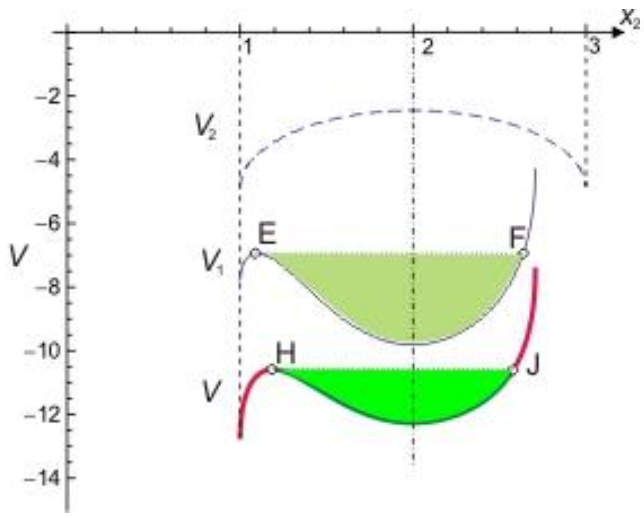

Fig. 4 *Potential wells for the masses and for the FPM as a whole for* $m_1 = 1$ *and* $m_2 = 0.25$

4. ANALYSIS OF THE EQUATIONS OF MOTION

4.1 Numerical analysis of the exact equations of motion

Equations (2-4) and (14-17) form the system of differential-algebraic equations (DAEs) of index 1. As it is known, DAEs are generally more difficult to solve than the system of ordinary differential equations (ODEs). Here, their solution is obtained numerically with the task of identifying qualitatively different responses and depicting them in the parametric plane defined by the initial angular displacement of the upper pendulum ($\theta_1(0) \equiv \theta$) and m_2 (Fig. 5). Two types of the response are found: the green dots depict the desirable period response, which is presented in Fig. 6a-c and Fig. 7a, while the red dots depict the undesirable collapsing behaviour, which is shown in Fig. 7b.

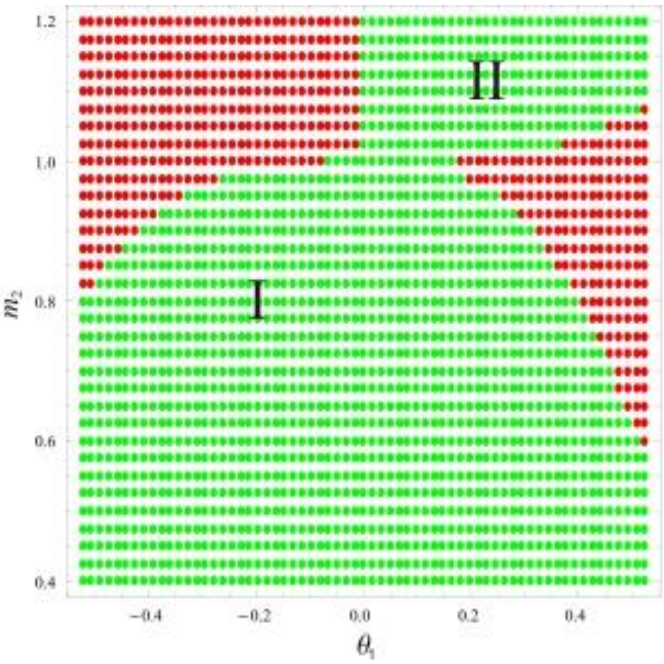

Fig. 5 *Parametric plane θ-m_2 plotted for m_1=1 with two qualitatively different types of the response labelled by the green dotes (desirable response) and the red dots (undesirable, collapsing response)*

The parametric plane presented in Fig. 5 can be divided into two regions with respect to the appearance of desirable motion: one corresponding to $m_2<m_1$ and the other one corresponding to $m_2>m_1$. They are respectively labelled by I and II and are in accordance with the potential well considered in Fig. 3.

To illustrate further the response obtained for the fixed value of m_2, but different angle θ, Fig. 6 and 7 are compiled, containing the time responses. Figure 6 shows the periodic response obtained for different values of the initial angular displacement: larger and positive (Fig. 6a), larger and negative (Fig. 6b), as well as small and positive (Fig. 6c). However, Fig. 7 illustrates that for the fixed m_2 but slightly different and small angle θ, a qualitatively different type of response can exist, which further implies the sensitiveness of the system to the initial conditions in this region.

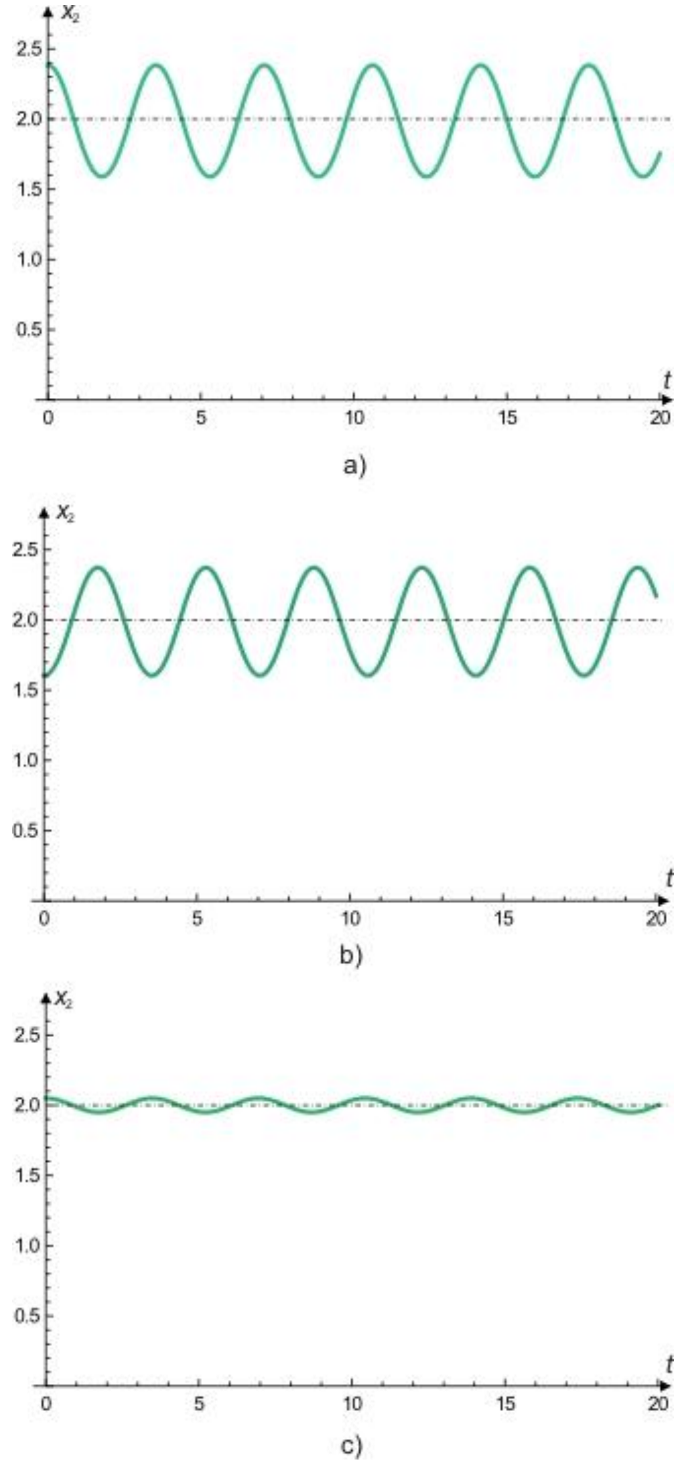

Fig. 6 *Time responses obtained numerically for m_2=0.5 and: a) $\theta = 0.4$; b) $\theta = -0.4$; c) $\theta = 0.05$*

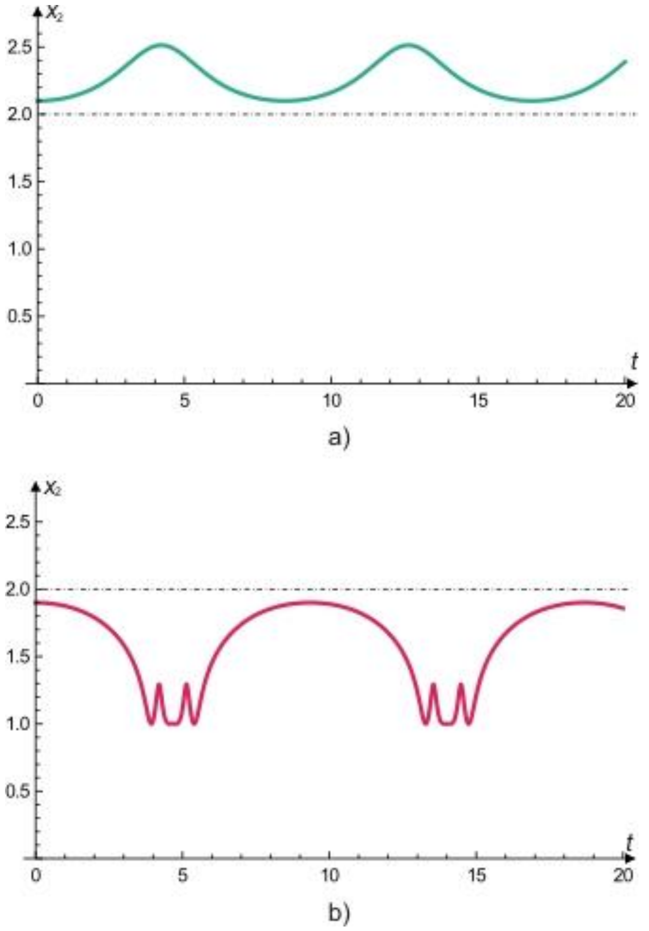

Fig. 7 *Time responses obtained numerically for m_2=1.1 and: a)* $\theta = 0.1$; *b)* $\theta = -0.1$

It should be noted that although the parametric plot and associated responses are shown only for one value of m_1 in Fig. 5, its structure remains the same, as confirmed in Fig. 8 plotted for m_1=0.5 and in Fig. 9 obtained for m_1=1.5.

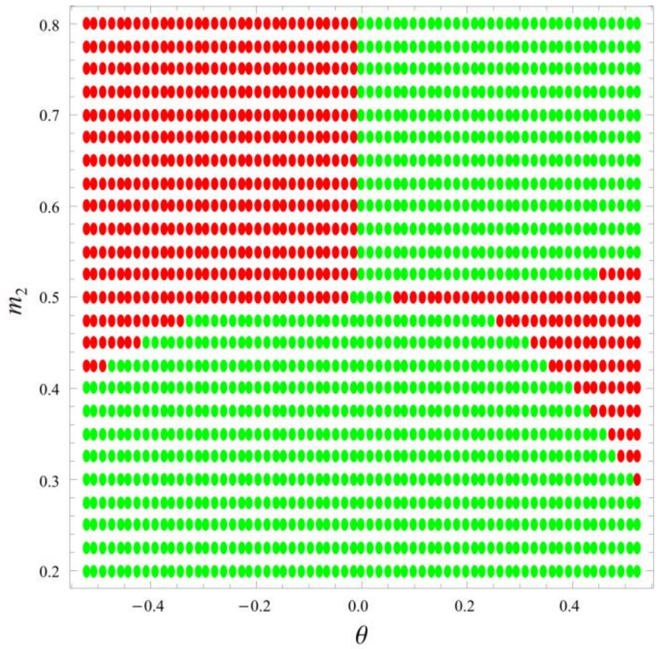

Fig. 8 *Parametric plane θ-m_2 plotted for m_1=0.5*

Namely, in the region m_2<m_1 (the lower horizontal half-plane), both types of the responses can exist depending on the combinations of m_2 and θ; in the region m_2>m_1 (the upper horizontal half-plane), the response is undesirable for θ<0, and desirable for θ>0. As far as the authors are aware, the existence of the desirable response in this region has not been known so far. What is its additional benefit is a considerably longer period (see Fig. 7a) than in the case shown in Fig. 6a-c.

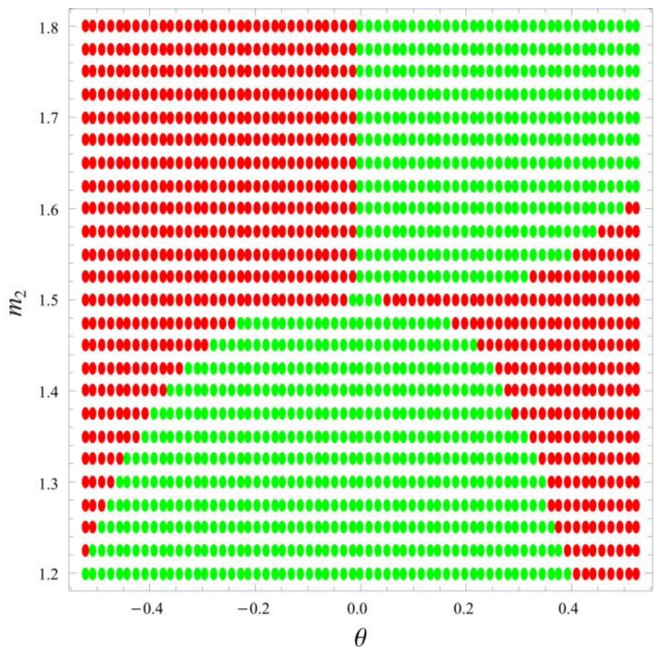

Fig. 9 *Parametric plane θ-m_2 plotted for m_1=1.5*

4.2 Analytical analysis of the approximated equation of motion

It is assumed now that the system performs small oscillations around the stable equilibrium in which the bars OA and BD are vertical. The kinetic energy (5) can then be formed while considering the system passing through the stable equilibrium position, i.e. (see Fig. 2)

$$T = \frac{1}{2} m_1 l_1^2 \dot{\theta}_1^2 + \frac{1}{2} m_2 l_2^2 \dot{\theta}_2^2. \tag{22}$$

The potential energy (6) can be represented and approximated as follows

$$\begin{aligned} V &= -m_1 g\, l_1 \cos\theta_1 + m_2 g l_2 \cos\theta_2 + const. \\ &\approx m_1 g\, l_1 \frac{\theta_1^2}{2} - m_2 g l_2 \frac{\theta_2^2}{2} + const. \end{aligned} \tag{23}$$

Using Fig. 2, one can write down the following geometric constraint

$$l_1 \sin\theta_1 + l_3 \cos\theta_3 - l_2 \sin\theta_2 = l_3. \tag{24}$$

Linearising it, one can derive the relationship between the angles and identify it as follows

$$l_1 \theta_1 = l_2 \theta_2 \equiv x. \tag{25}$$

By introducing Eq. (25) into Eqs. (22, 23) and by using Lagrange's equation of the second kind (see Eq. (4) with by λ_j =0) with the generalised coordinate being the new coordinate *x*, one can derive

$$\ddot{x} + \frac{g}{m_1 + m_2}\left(\frac{m_1}{l_1} - \frac{m_2}{l_2}\right)x = 0. \tag{26}$$

The form of this equation is the same as the one for a linear (simple harmonic) oscillator, so that the natural frequency can be recognised as

$$\omega = \sqrt{\frac{g}{m_1 + m_2}\frac{m_1 l_2 - m_2 l_1}{l_1 l_2}}. \tag{27}$$

Equation (27) implies that the natural frequency of small oscillations depends not only on the mutual relationship between the masses m_1 and m_2, but also on the geometry of the mechanism, i.e. on the lengths l_1 and l_2. Moreover, in order for such oscillations to be achievable, the following condition must be satisfied

$$m_1 l_2 > m_2 l_1. \tag{28}$$

It should be emphasized that this result is related to the potential well HJ (Fig. 4) and the region I in Fig. 5. However, this approximate approach does not cover the region II in Fig. 5 and it points out the necessity to carry out further nonlinear analysis.

5. CONCLUSIONS

This study has been concerned with a folded pendulum mechanism. Its motion is subjected to certain geometric constraints, which is the reason why the equations of motion have been derived in the form of Lagrange' equations of the first kind with undetermined multipliers. The resulting system of differential-algebraic equations has been solved numerically to detect characteristic responses and they have been related to the values of the system parameters. As far as the authors are aware, these results are new. The comparison with the linearised equation of motion has revealed that this approximation just partially covers the potentially beneficial behaviour. The numerical analysis has shown the existence of the desired response in the region out of the domain covered by the linear one. This response even has a significantly longer period. These findings also pointed out the necessity of further nonlinear investigations.

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REFERENCES

[1] D.G. Blair, J. Liu, E.F. Moghaddam, L. Ju, "Performance of an ultra low-frequency folded pendulum", *Physics Letters A*, vol. 193, pp. 223-226, 1994.

[2] J. Liu, L. Ju, D.G. Blair, "Vibration isolation performance of an ultra-low frequency folded pendulum resonator", *Physics Letters A*, vol. 228, pp. 243-249, 1997.

[3] J. Liu, D. G. Blair, L. Ju, "Near-shore ocean wave measurement using a very low frequency folded pendulum", *Measurements Science and Technology*, vol. 9, pp. 1772-1776, 1998.

[4] S. Fan, Y. Cai, S. Wu, J. Luo, H. Hsu, "Response of a folded pendulum to tilt tides", *Physics Letters A*, vol. 256, pp. 132-140, 1999.

[5] A. Bertolini, R. DeSalvo, F. Fidecare, M. Francesconi, S. Marka, V. Sannibale, D. Simonetti, A. Takamori, H. Tariq, "Mechanical design of a single-axis monolithic accelerometer for advanced seismic attenuation systems", *Nuclear Instruments and Methods in Physics Research*, vol. 556, pp. 223-226, 2006.

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HYDROSTATIC SYSTEMS FOR VIBRATION DAMPING IN THE MOVEMENT OF MOBILE MACHINERY

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Abstract - *The paper discuss the concept of variant hydrostatic systems for vibration damping in the movement of mobile machines. The systems have been developed on the basis of a mathematical model of mobile machinery in the form of dynamic absorber consisting of a support-moving member with tires, elastically leaning on a path of movement, and manipulators that are elastically connected to moving member with the hydraulic cylinder that acts like "hydraulic springs". Vibration damping in the movement of machines is based on the possibility of changing the volume of hydraulic oil in the hydro-cylinder ports when the manipulator is lifting and lowering. The change of oil volume is achieved by installation of special modular components in the hydrostatic drive system of the machine while dynamic characteristics of the machine are changing thus stabilising the vibration of movement.*

1. INTRODUCTION

The basic function of the loader has the following partial functions - operations: loading F_1 (Fig.1a), transport F_2, unloading F_3 of materials and again return to a new place of loading F_4. The cycle starts by selection of the loading workspace. A primary function of the loader is cyclic transport of bulk material to a certain distance (*max*. to $l=50$ *m*).

The loading operations can be achieved in different ways that are adjusted to the type and configuration of the diposed material. After loading, the material is raised to the required height above the supporting surface and transpored to the unloading place. During the transport, and approaching to the unloading space at the same time is done raising of materials on certian dump height. Unloading is doe in a pile or in a trailer (bucket) of transport equiment.

Basic function of loader of all sizes, is realized by general configuration of the kinematic chain consisting of: a back part L_1 (fig. 1a) and the front part L_2 of the support-moving mechanism and manipulator with an arm L_3 and bucket L_4. Support-moving mechanism is mainly performed on the tires. An actuator C_3 of the arm's drivetrain mechanism are two hydro-cylinders of bidirectional effects, which are directly conected to the support-moving mechanism and arm. An actuator C_4 of bucket's drivetrain mechanism is hydro-cylinder of bidirectional effects and it is indirectly conected to support-moving mechanism and bucket with "Z-working mechanism". The hydro-cylinders of powertrain mechanism are powered with hydraulic pumps in an open hydraulic circuit. The analysis of functions shows that the operation of transport is largest in the cycle duration, and it is achieved with different paths of the movement of loader, depending on site of loading and unloading of materials. It is characteristic that the path of the movement is usually achieved on a unarranged terrain. At the transport operation wheel loaders are moving with a full and at return with an empty bucket [1].

The speed and the mass to be transported are decisive for high economy. In this respect, wheel loaders struggle with the disadvantage that the bucket with the load acts as a mass around the pivot point of the front axle.

Owing to the bucket mass and the distance between the bucket and the front wheel, which has the effect of a tilting fulcrum, wheeled loaders are particularly susceptible to rocking motions, particularly as they have no shock absorer system between the wheels and the chassis such as the dampers.

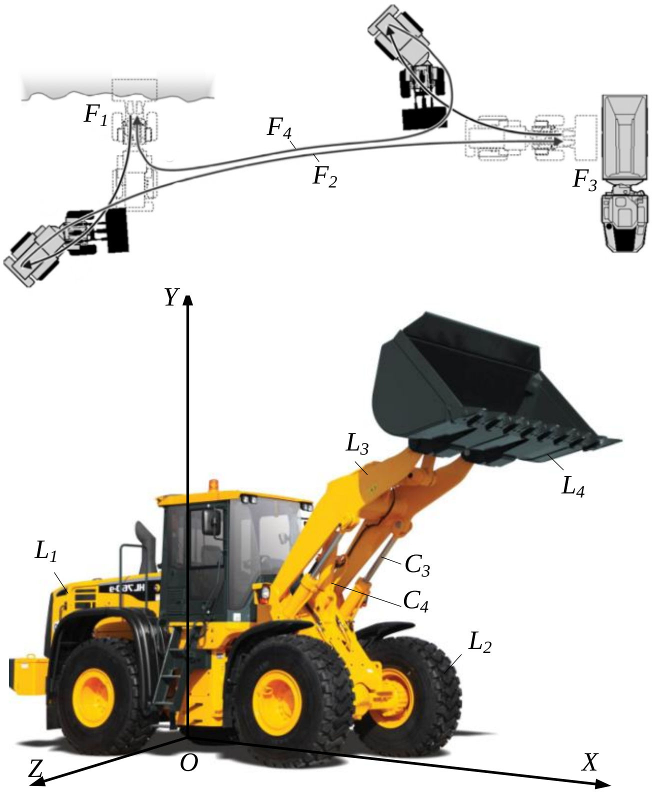

Fig. 1. *a) Functions and b) physical model loader*

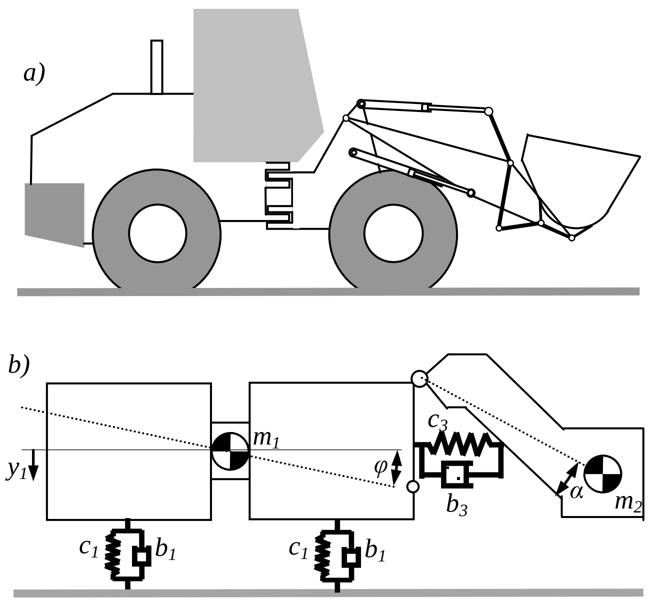

Fig. 2. *Models of the wheel loaders: a) physical, b) mathematical*

For technical reasons, they have instead only a limited wheel damping capability via the tyres. One effect of this is that, depending on the load and ground conditions, rocking oscillations can occur. The vehicle, the load and the driver are all exposed to these sometimes heavy movements. Aside from the mechanical stress placed on the machine, which can greatly increase wear in the long term, it has also been proven that this motion also harms the health of the driver. Other consequences of rocking include reduced handling efficiency, increased braking distances, impaired steering response and lower transportation speeds [2][3].

2. DYNAMIC ANALYSIS

In aim to analyze the oscillatory behavior of the loader's movement it has been developed different dynamic mathematical models. Where the loader's physical model is observed as a system with two rigid bodies: the moving mechanism and the maniplator with the following assumptions (Fig. 2a) [4]:

- members of the loader's kinematic chain and the loader's supporting surface are rigid bodies,
- tires and hydro-cylinders of the manipulator's drivetrain are elastically-damping elements,
- tires has point contact with the ground,
- continuous tires-ground contact,
- linear movement of the loader,
- the center of mass of the support-moving member and manipulator lie in a longitudinal vertical plane.

Generalized coordinates of the system are: vertically displacement y_1 of the support-moving mechanism (Fig. 2b), the angle of inclination φ of the support-moving mechanism and the angle α is displacement of manipulator around the horizontal axis of the linkage, which is linked to the support-moving mechanism.

Seted dynamic mathematical model of the wheel loader is similar to dynamic mathematical model of the absorber. In principle, the dynamic absorber consists of two masses: m_1 - in the case of the loader that is the mass of the support-moving mechanism and m_2 mass absorber or mass of manipulator. The masses are connected with elastically-damping elements of specific stiffness c_i and damping b_i that depend on the characteristics of tires, geometry of hydro-cylinders of the loader's manipulator, volume and characte-ristics of the hydraulic oil in their working working ports. An aim of the dynamic absorber is that the specific characteristics of elastic-damping elements leads to reduction of the amplitude of mass m_1, apropos support-moving mechanism on which there is a cabin and in it is machine's operator.

3. MOTION REGULATION

When wheel loaders move with velocity higher than *10* [*km/h*], they are subject to intense longitudinal vibrations. These vibrations are experienced particularly unpleasantly by an operator. Contemporary wheel loaders, which are required to move relatively fast reaching the speed of *60* [*km/h*], are often subject to intense longitudinal vibrations.

By using the effect of the dynamic absorber it has been developed an active and passive regulation of motion systems of the loader in order to decrease the vibration that act on the machine's operator.

The active regulation systems of the loader's movement are based on a change in the flow of hydraulic pump that powers the drivetrain of the manipulator's hydro-cylindres. By changing the flow of hydraulic pump in same time the elastic-damping characteristics of the manipulator's hydro-cilinders are changing so that the vibration of the loader's support-moving mechanism is reduced.

The change of hydraulic pump's flow depends on the size of the following loader's state variables.

- vertical displacements y_1 and velocity of the suport-moving mechanism,
- angle φ and angular velocity of inclinaition of the support-moving mechanism,
- the angle α and the angular velocity of displacment of the manipulator,
- pressure on the piston's side of the hydro-cilinder of the arm's drivetrain mechanism.

The passive regulation systems presents hydraulic amortization by using elastic-damping connections of the manipulator for the support-moving mechanism. Whereby, the manipulator of machine is using like amortization mass of support-moving mechanism. The passive systems enables flexible upgrade of the wheel loaders where result is reduction of osclillation motion of the machine apropos increasing of operater's comfort and safety. The passive systems have possibility of adjusting the characteristics and turning off the system when the loading operation is active.

As an example, here is given scheme of hydrostatic system (Fig.3) of the manipulator's drivetrain with the passive hydraulic stabilisation module of two different varinats.

The first variant relates to the stabilisation modul for larger wheel loaders with a nominal flow of *300* [*l/min*], and the second variant relates to the stabilisation modul with a nominal flow of *80* [*l/min*], for smaller wheel loaders.

The hydrostatic system of manipulator's drivetrain of the wheel loader contains of: main hydraulic pumps (1) which are powering via valves (3) the actuators of the manipulator: the hydro-cylinders of arm (5) and the bucket (6). The boost pump (2) powered the pilot valves (4) and the stablization module (7) with accumulators (8). The manipulator's operating is realized by pilot valve (4.1) which actived movement of the arm's hydro-cylinders (5) and by pilot valve (4.2) movement of bucket's hydro-cylinder (6) is actived.

During driving the stabilising module (7) is activated via solenoid valve (7.2) dependent on the driving speed (between *4* and *7 km/h*) and thus the arm's cylinder cap side is connected to the hydraulic accumulators (8).

At the same time, solenoid (3.1) in the control block (3) is operated for switching the stabilisation module spool, which connects the piston rod side to the tank. This ensures cavitation-free oscillating of the arm's cylinder

Variant I - The stabilising module for larger wheel loaders basically comprises of a housing into which are built:

- Valve spool (7.1)
- 3/2-way directional valve, solenoid operated (7.2)
- Pressure relief valve (7.3)
- Emergency drain screw (7.4)

If the lifting cylinder (5) has pressure applied to the piston side, then the pressure is also applied to the check valve in the valve spool (7.1) and the accumulator (8).

Dependent on the design the connection from the lifting cylinder (5) to the accumulator (8) via the valve spool (7.1) is interrupted (switched position 2).

A pressure reducing function for the accumulator (8) is integrated in the valve spool (7.1) (switched position 3). The opening pressure lies approx. 30 bar higher than the switch off pressure (switched position 2).

The damping valve can be automatically activated via the travel speed. The 3/2-way directional valve (7.2) is switched into the switched position 2. The valve spool (7.1) is switched to the switched position 4 and connects the piston side of the lifting cylinder (5) with the accumulator (8) as well as the rod side of the lifting cylinder (5) with the reservoir.The pressure relief valve (7.3) prevents unpermissible high pressures in the accumulator (opening pressure < permissible accumulator pressure).

Variant II - The stabilising module for smaller wheel loaders basically comprises of a housing into which are built:

- Valve spool (7.1)
- 3/2-way directional valve, solenoid operated (7.2)
- Pressure relief valve (7.3)
- Emergency drain screw (7.4)
- Accumulator loading valve (7.5)

The difference of variant I with respect to a variation II is in the fact that the loading and unloading speed of the accumulator is defined via the selectable orifice cross-sections on the accumulator loading valve (7.5) [6].

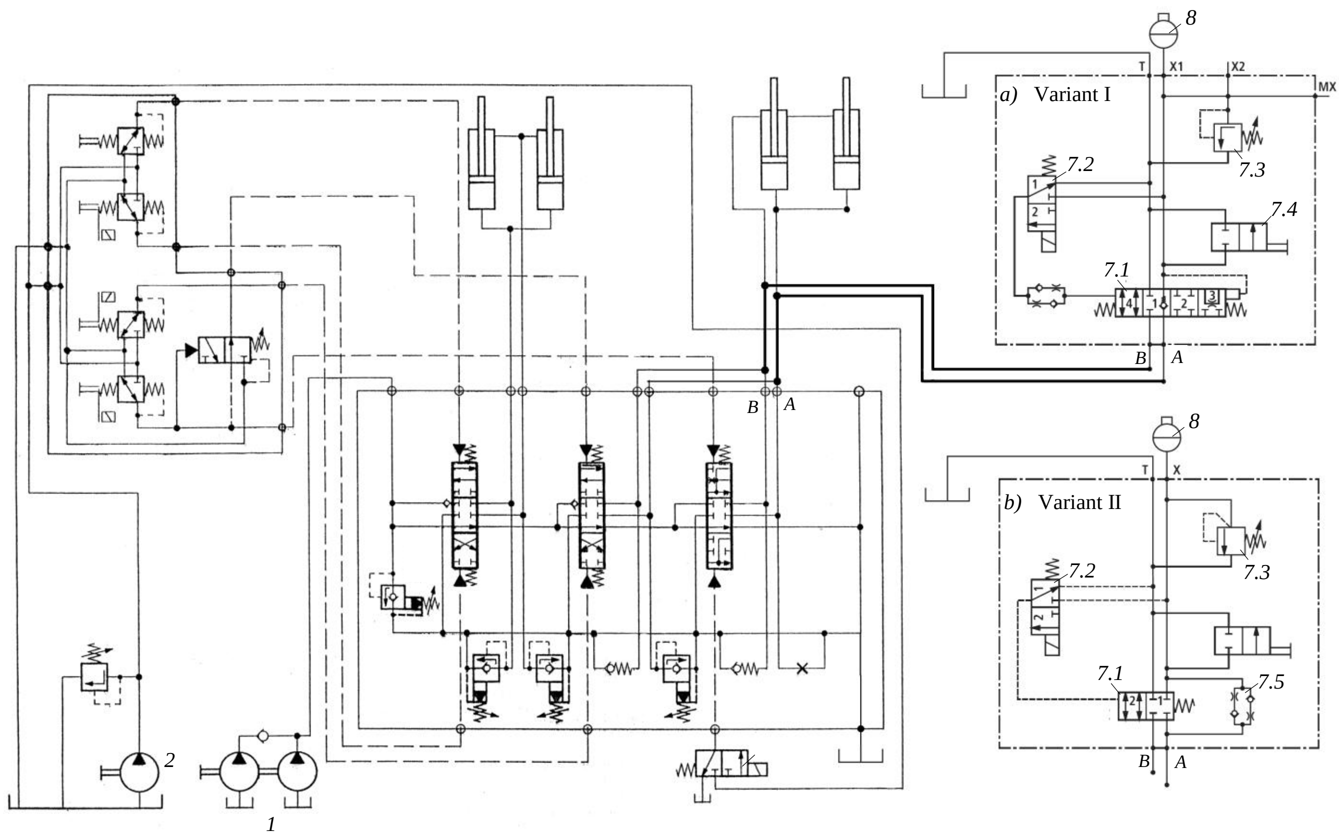

Fig. 3. *Functional scheme of the wheel loader's hydrostatic system with: a) stabilisation module variant I, b) stabilisation module variant II*

As the pressure in the accumulator always equals or is higher than that in the arm's cylinder, the bucket is always slightly raised and cannot drop during driving when the stabilising system is connected. The stabilising system improves the driving characteristics irrespective of whether the bucket is empty or loaded.

The main elastic-damping characteristiks of the stabilization module depend on accumulator (8). The elastic-damping characteristiks of the accumulator are determined depened on state values of pressure and volume of filling of accumulator's gas.

Filling the accumulator presents termodynamic process where is heat changed between the hydraulic accumulator and an enviroment by a law of politrope, which is expressed with equation:

$$p_i \cdot V_i^n = const \tag{1}$$

Based on previous equation it can be determined elastic characteristiks of the hydraulic accumulator:

$$c = \frac{n \cdot p_i \cdot A_k^2}{V_i} \left(\frac{1}{1 - \frac{A_k}{V_i} \cdot s} \right)^{n+1} \tag{2}$$

where is: n - exponent of politrope, p_i - pressure of system, A_k - area of piston, V_i - volume of inert gas, s - motion of piston, c - stiffness of accumulator.

At the hydraulic accumulator's construction must pay attention that between arm's hydro-cylinder and hydraulic accumulator short ports with big nominal diametar is seted.

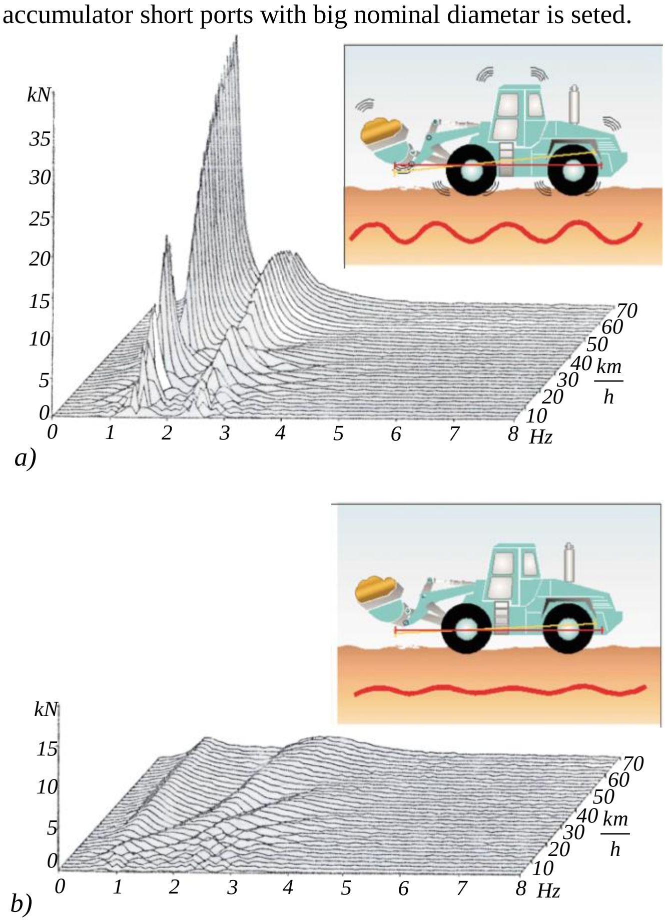

Fig.4. *Loads of loader's front axle: a) without stabilization module, b) with stabilization module [7]*

It has been done researhes of the wheel loaders with and without stabilization module at difference condition of movement in term off the terrain configuration and the speed of movement with the loaded and empty manipulator's bucket.

For the comparative evaluation of behaviour at loader's movement it has been measure on the front and rear axles on moving mechanism: a vertical loads (Fig.4), seat's acceleration, motion of piston of arm's hydro-cylinder and speed of movement.

The results of researhes, shows that with stabilization module, dynamic's loads of axles of loader's moving mechanism are less from 40% to 65%, while seat's acceleration are less from 48% to 60%. A spectrums shows that at the largest value of axle's load are at frequency: $f=1.64$ [Hz] and $f=2.68$ [Hz] [7].

4. CONCLUSION

When the wheel loaders are doing their manipulative task and special at the operation of transport from loading site to site of unloading of the materials, they acting like very sensible dynamic systems with appearance of disadvantage oscillations on control site of the machine's operater.

From that reason the researches are done with aim to appease oscillations at loader's motion and to reform ergonomic conditions of the maintenance. Using the principle of dynamic apsorber, it has been developed stabilization modules which construction in present hydrostatic systems of the loaders, can be done successful motion stabilization and comfort of maintenance.

Principaly, the stabilization modules content the hydraulic accumulator with membrane, which stiffness in dynamic system of the manipulator, is damping oscillations of the support-moving mechanism, apropros the operator cabin.

ACKNOWLEDGMENT

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REFERENCES

[1] D. Janošević, "Projektovanje mobilnih mašina", *Faculty of Mechanical Engineering University of Niš, ISBN 8680587443*, Niš, 2006.

[2] H. Fischer, "Open-Loop and Closed-Loop Control Concepts in Wheeled Loaders", *Proc. International Mobile Conference*, Ulm, Germany, pp. 67-78, 2000.

[3] M. Frasch, D. Seubert, H. Funk, E. Harnischfeger, T. Krumrey, U. Sonnenburg and M. Schulte, "Fit for the Future – New Developments for Wheel Loaders", *Proc. International Mobile Hydraulics Conference in Ulm*, Germany, pp. 70-93, 2003

[4] P. Bratu, "The Analysis of the Dynamic Loads Due to the Vibrations Generated by the Displacement of Front Loaders", *IMK-14 – Research & Development*, vol. 18 (2), pp. 37-42, 2012

[5] A. Kosiara, "Control Effectiveness Comparison of Wheel Loader's Longitudinal Vibrations by Means Of Passive and Active Vibration Stabilisers", *Journal of KONES Powertrain and Transport*, Vol. 18, No. 1, pp. 287-294, 2011.

[6] Rexroth Bosch, "Stabilising Module", Type RSM2, Product catalogue, 2004.

[7] W. Poppy, "Hydraulische Tilgung betriebsbedingter Schwingungen bei selbstfahrenden Arbeitsmaschinen", *Journal Construction vol. 38*, No. 12, Springer-Verlag, pp. 461-468, 1986.

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VIBRATIONS OF COMPOSITE BRIDGE STRUCTURES EXPOSED TO ACTION OF MOVING LOAD FROM VEHICLES - CASE STUDY

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Abstract - *This paper presents problems of dynamical interaction between a vehicle and a bridge. Results of in situ testing on composite steel-concrete bridges are used for dynamical response analysis. Particularly, impact factor from measured deflections and FFT analysis are evaluated, considering different traffic conditions.*

1. INTRODUCTION

Dynamical influence on elastic structures, and particularly the influence of moving load on bridge structures, represents a very old, and at the same time a very complex problem. Due to a large number of influent parameters the designers resort to avoidable idealizations, which, in particular and cumulatively affect the accuracy of the solution.

In many papers from this field, in attempt of explaining of multi-parametric influences, two different methods were treated: experimental methods and pure theoretical investigations. In this paper were analysed experimental investigations of the dynamic response of composite bridge structures of steel and concrete in case of excitation of dynamic action from moving (heavy) vehicle. Such type of excitation represents a real case of exploiting load and has practical application in testing of road bridges "in situ".

Regarding the complexity of the problem and wide spectrum of possible situations during testing, regulations for bridge testing (JUS U. M1. 046) do not have a precisely defined procedure of examinations of dynamic characteristics, neither regarding the type of load, nor regarding the load phases, but the testing programme is made for every single case.

An unavoidable, often crucial, parameter is undoubtedly the economical effect, that is, the cost of the performed testing, which unfortunately may lead to undesired improvisations.

This is an author’s attempt that, based on literature insight into the mentioned problem and long-life experience, on concrete examples, provide a possible procedure of an optimal range of testing and present possibilities of processing of obtained data using modern equipment (SPIDER 8) and adequate software packages (CATMAN).

2. DYNAMIC INTERACTION VEHICLE-BRIDGE

Dynamic interaction between a vehicle and a bridge is a very significant and complex problem which depends on series of parameters connected to the dynamic characteristics of the vehicle, dynamic characteristics of the bridge, and on relation between them.

Dynamic characteristics of the vehicle are function of the following parameters: number of axles, load per axle, axle distance, suspension, mass of the vehicle, speed, natural frequency, etc.

Dynamic characteristics of the bridge depend on: dimensions and geometry, types of restraints, distribution of mass and stiffness, cross-section and side section, damping, unevenness on the bridge and in front of it, etc.

Relations between dynamic characteristics of the vehicle and the bridge apply on: mass coefficient (relation of the vehicle mass and the bridge mass), mutual frequencies and oscillation period.

As one may see, a complete dynamic analysis requires knowing of very large number of parameters. Because of that, and due to practical reasons, a simplified relation between dynamic and static characteristics was used in design of bridges, using an amplification factor (dynamic coefficient, or impact coefficient), whose value in corresponding national codes depends mainly from the bridge span, and it is defined rather coarsely.

AASHTO standard, for example, in its previous version, for dynamic calculation takes static values amplified by impact factor (IF). The new AASHTO takes also the LRFD (load resistance factor design) factor for the calculation of resistance to load. The second possibility is determining of the DAF (dynamic amplification factor), with the following relation:

$$DAF = 1 + \left(\frac{\delta_{din} - \delta_{stat}}{\delta_{stat}} \right) \tag{1}$$

This expression has similar form in many standards, and in our country is defined as:

$$1 + \varphi = 1 + \left(\frac{Y_{\max} - Y_{\min}}{Y_{\max} + Y_{\min}} \right) \tag{2}$$

3. EXPERIMENTAL INVESTIGATIONS

3.1 Example of the bridge over Grošnička river in Kragujevac

An investigation of a bridge with a span of 26,8 m (Fig. 1), and with static system of a simply supported beam, with 5 main girders of variable depth, compound with a cast-in-situ concrete deck by slender dowels, was conducted (Fig. 2).

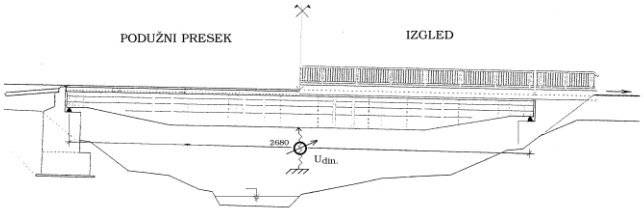


Fig. 1 *Elevation and longitudinal section of the bridge*

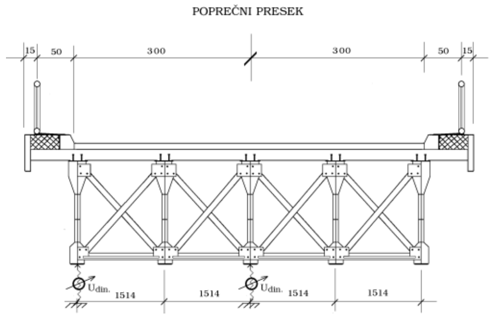


Fig. 2 *Cross section of the bridge*

Examination programme of the dynamic characteristics of the bridge encompassed (depending on the available number of measuring channels) simultaneous tracking of a large number of measuring spots, at different load constellations.

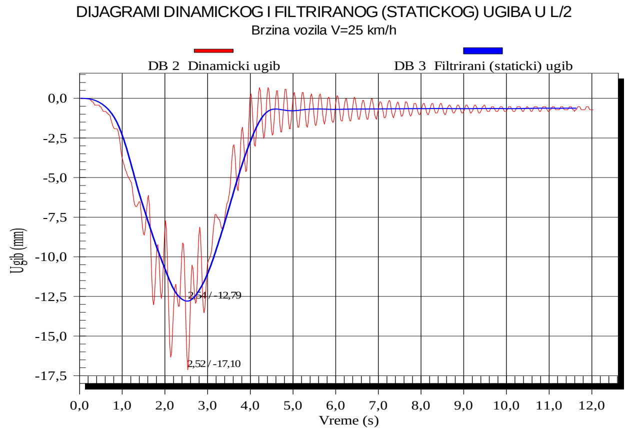
The measuring spots were designated for tracking of the dynamic deflection by LVDT at the midlength of the bridge, on the middle and on the edge girder. There were also measured the response of the bridge by acceleration gauges on the main girders, at ½ and ¼ of the bridge span, and dilatations in the concrete deck and in the dowels.

Load constellations encompassed: bridge crossing with different velocities, crossing over an obstacle at the midspan, braking in the midspan, crossing of two trucks (one after another), braking over supports.

For the lower frequency ranges (1÷10 Hz), where the bridge structures belong too, more favourable is an analysis of the dynamic coefficient based on the measured deflection, using a LVDT, than the analysis based on acceleration, obtained by acceleration gauges.

For these reasons, as well as for the limited space, we analysed only the response of the bridge based on the measured deflections. For different load constellations and different structural elements a dynamic (impact) coefficient was determined and FFT analysis was done. Digital filtering of the signals was used to define a quasi static deflection, generated by action of the same moving vehicle, in order to define the real dynamic coefficient in the most precise way.

3.1.1 Results of deflection measuring on the middle main girder in the midspan



Value of the dynamic (impact) coefficient may be obtained as:

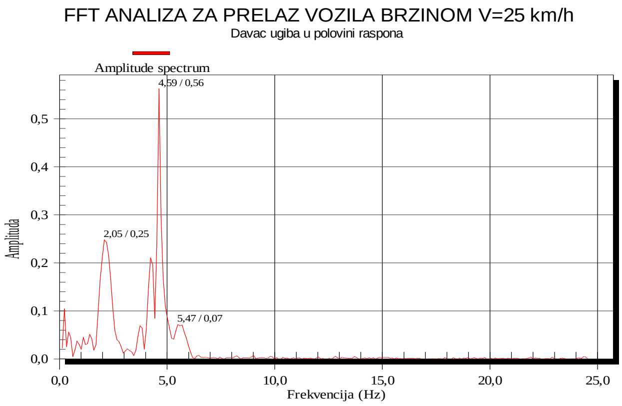
$$DAF = 1 + \left(\frac{\delta_{din} - \delta_{stat}}{\delta_{stat}} \right) = 1 + \left(\frac{17{,}1 - 12{,}79}{12.79} \right) = 1.337 \qquad (3)$$

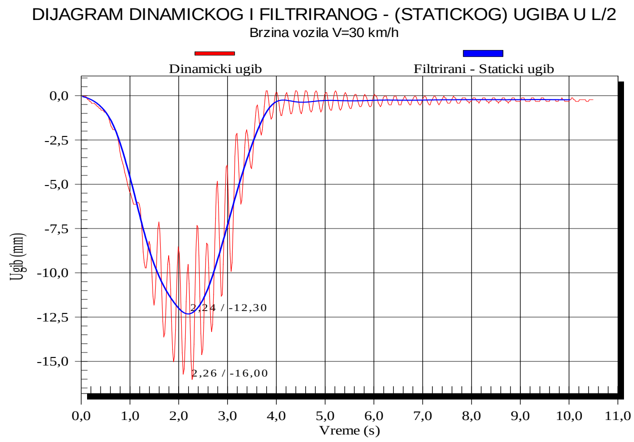
or directly:

$$DAF = 1 + \varphi = \left(\frac{Y_{din}}{Y_{stat}} \right) = \frac{17{,}1}{12{,}79} = 1{,}337 \qquad (4)$$

Dynamic coefficient was determined for the remained load constellations too, and presented in the tables.

The FFT analysis was conducted for the amplitude-phase spectrum at different load constellations.





FFT ANALIZA ZA PRELAZ VOZILA BRZINOM V=30 km/h

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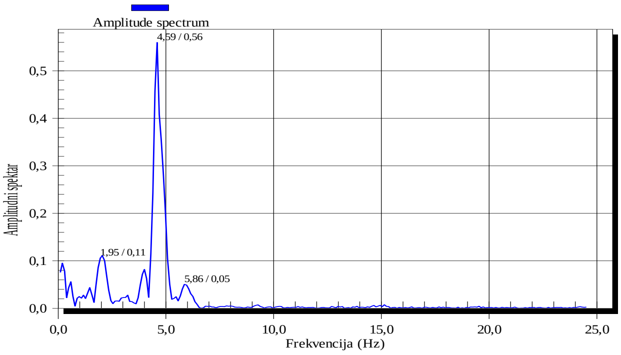

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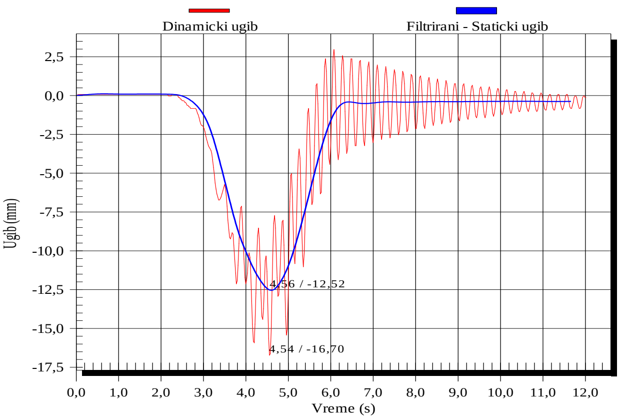
Brzina vozila V=35 km/h



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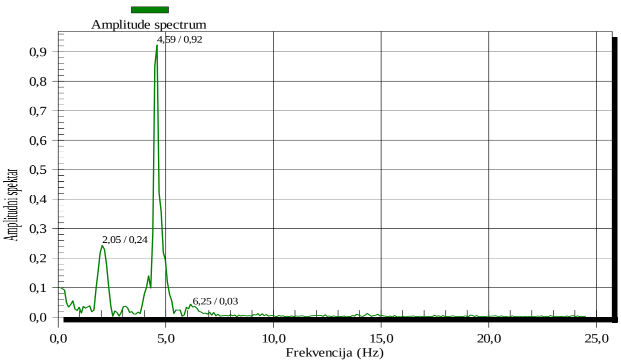

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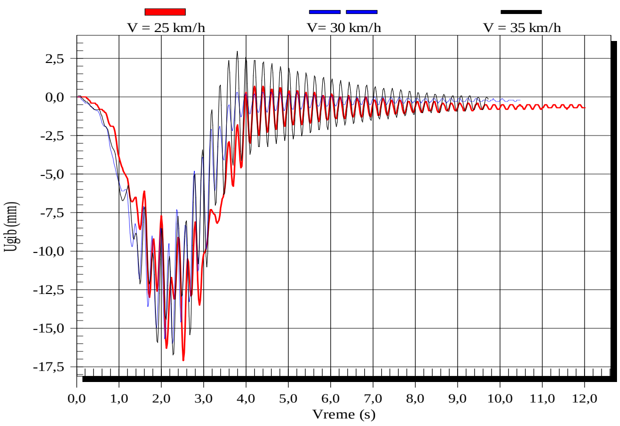

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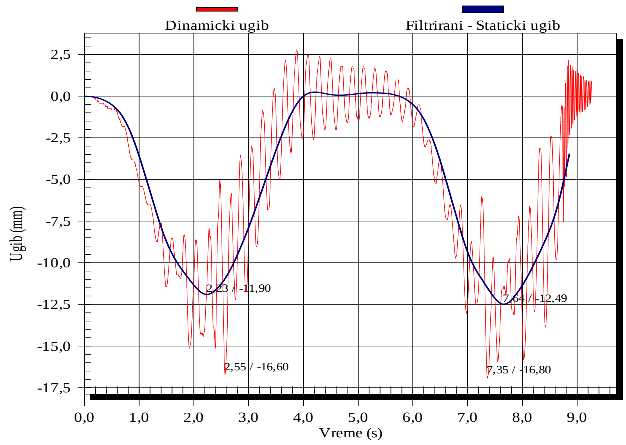

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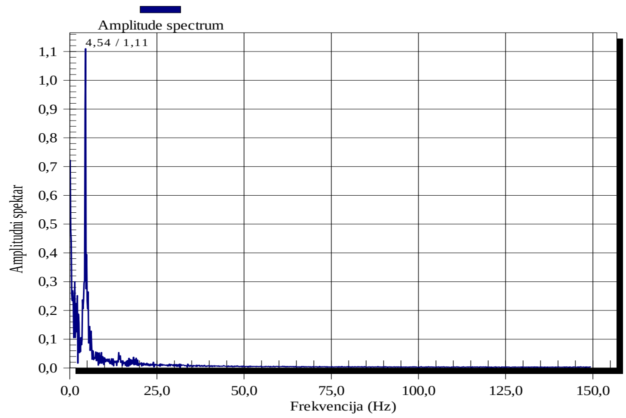

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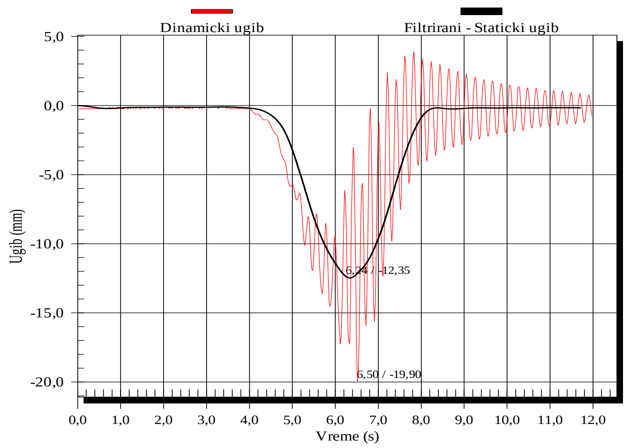

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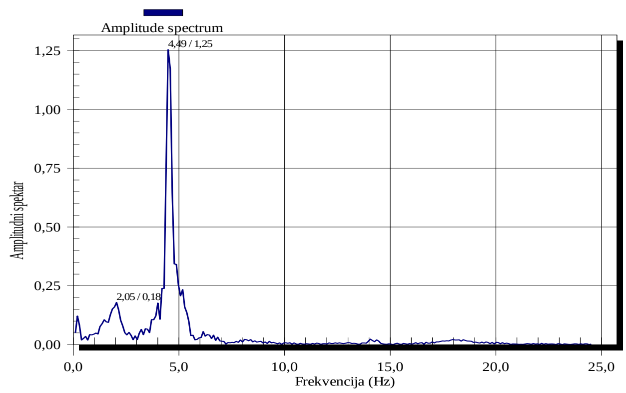

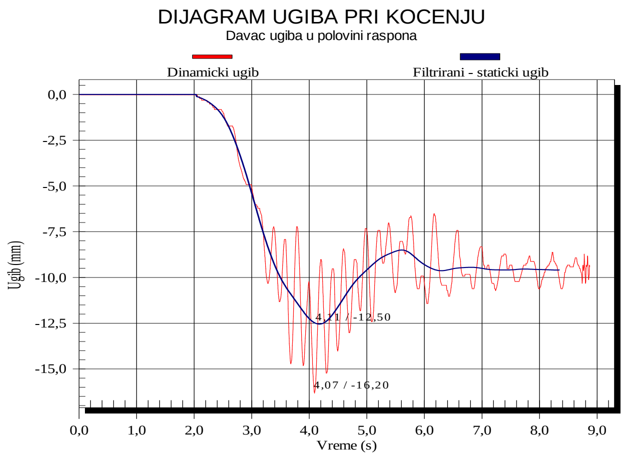

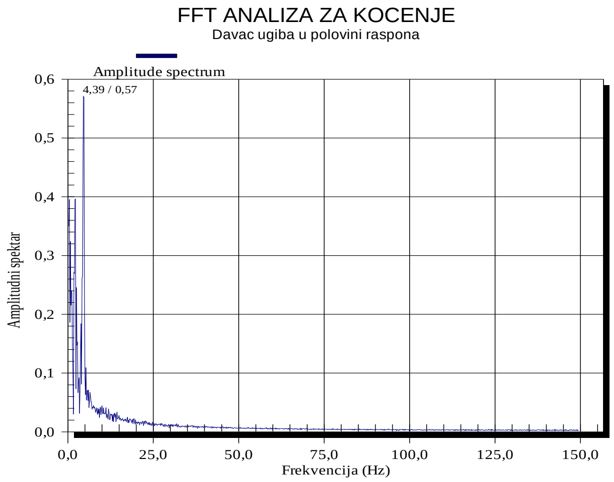

In the Table 1 are presented values of the measured deflections (dynamic and filtered – quasi static), calculated dynamic coefficients, and frequencies for different load constellations.

Table 1 *Dynamic parameters*

Load Constellation	Ydyn (mm)	Yfilt. (mm)	Dyn. Coeff.	Frequency (Hz)	Amplitude spectrum
V=25km/h	17.1	12.79	1.337	4.59	0.56
V=30km/h	16.0	12.30	1.301	4.59	0.56
V=35km/h	16.7	12.52	1.334	4.59	0.92
Obstacle	19.9	12.32	1.611	4.49	1.25
Braking	16.2	12.50	1.296	4.39	0.57
2 vehicles	16.6	11.90	1.395	4.54	1.11
	16.8	12.49	1.345		

4. CONCLUSION

Having in mind the significance of the dynamic coefficient for rational design of the bridge structures, the need for its as accurate as possible determination is intruded. At composite structures this problem is additionally complicated because two materials with different characteristics and stiffness exist, and new structural elements emerge – dowels. Because of that, dynamic increase of deflection, bending moment, shear force and reactions often is not the same, neither is the same the value at specific structural elements of the composite section (girder, deck, dowels).

Dynamic coefficient is experimentally obtained in the most convenient way based on measured deflections, dynamic and static. A very fast and efficient technique of digital filtering of the record of the dynamic deflection is applied, in order to determine the quasi static deflection from the same load as a function of time.

The data obtained by experiment show (Table 1), that dynamic (impact) coefficient often has not a direct correlation with the vehicle speed, that its increase is substantial in case of setting an obstacle on the bridge floor in the form of a plank with depth of 5 cm, and that at braking it keeps similar value as at truck crossing.

The FFT analysis showed a negligible difference of the frequency of the basic oscillations (4.39 ÷ 4.59 Hz) at change of speed, i.e., load constellation (crossing, obstacle, braking).

One may notice a direct relation between the amplitude spectrum and dynamic coefficient, i.e. grow of the amplitude spectrum with the increase of the dynamic coefficient.

Unevenness on the roadway which accompany our bridges in exploitation, have significant influence on dynamic behaviour, and by that on the durability of the bridge, which may and must be investigated practically by setting an artificial unevenness (obstacle) on the bridge.

For the final assessment about dynamic behaviour of a bridge structure, it is obviously necessary to analyse larger number of load constellations, and at the same time to track several parameters. Here, two important parameters were specially treated: dynamic coefficient and frequency.

Based on the obtained results, one may conclude: 1) due to the relatively small variation of the dynamic coefficient as a function of speed, a single crossing of the vehicle at maximal speed is sufficient for the concrete traffic conditions; 2) it is mandatory to set an artificial unevenness (obstacle) at the location of the maximal influences, because in that case the dynamic coefficient increases significantly; 3) influence of braking on the bridge has greater significance for defining of damping than for the dynamic coefficient or frequency; 4) Behaviour of the dowels has specific dynamic characteristics, often different than those regarding the bearing structure, so they should be investigated obligatory.

REFERENCES

[1] J. T. Broch: *Mechanical Vibration and Shock Measurments*, 1980.

[2] P. Paultre, J. Proulx, M. Talbot: *Dynamic Testing Procedures for Highway Bridges Using Traffic Loads*, Journal of structural engineering.

[3] O. S. Salawu, C. Williams: *Review of full-scale dynamic testing of bridge structures,* Eng. Struct. 1995., Volume 17, Number 2.

[4] *Izveštaj o ispitivanju drumskog mosta preko Grošničke reke u Kragujevcu*, Institut GAF 2012.

THE EVALUATION OF TRACTOR SEAT CUSHION MATERIALS USING THE ANALYTIC HIERARCHY PROCESS

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Abstract - *While performing agricultural operations, a large number of tractor drivers are faced with the lack of even minimal oscillatory comfort. Older tractors are not able to, during the ride and under different conditions, reduce the oscillation impact of certain construction parts to a man to the minimum. The number of potential solutions to this problem in such tractors is almost reduced to a minimum, partly due to technical reasons (feasibility), and partly due to economic reasons (high price). One technically simple and economically justified solution is based on the fact that some materials or fluids can isolate the vibrations in a better way and can provide a greater comfort to the driver than the regular sponge, which is a typical material for seat coating in older tractors. As possible alternatives to the original seat, the cushions filled with hard-pressed sponge, water and air are investigated in this paper. The multi-criteria analysis, the analytic hierarchy process method, was used to determine the quality of each cushion individually. Besides the general comfort of the cushion and its ability to reduce vibrations, the price of the pillow was also selected as a criterion. Depending on the importance, each criterion in the analysis can get the adequate importance criterion. The optimisation of oscillatory comfort was set as the primary objective in this paper, so the greatest importance was given to the vibration reduction criterion. The cushion filled with air turned out to be the best alternative, which had satisfactory results in the majority of criteria.*

Key words*: tractor, vibrations, multi-criteria analysis, analytic hierarchy process.*

1. INTRODUCTION

As one of the most common means of agricultural machinery, tractors have had a significant influence on the increase in productivity and efficiency of agricultural operations and on the direct relief, and in some cases even the complete elimination of physical labour of agricultural workers. On the other hand, in the course of their daily work activities, tractor drivers are exposed to numerous adverse influences which have a complex harmful effect on human health and efficient performance of duties. Apart from physical efforts, noise, atmospheric precipitation, high humidity, high or low temperature, dust and various chemical types of pollution, vibrations also appear as one of the harmful effects [1].

Vibrations appear when tractor's engine is turned on, in interaction with uneven terrains. In working conditions, the entire vehicle is exposed to complex oscillatory processes which are transferred from the engine through the transmission and chassis to the cabin and further, through the floor, seat and operating commands, to the body of the driver.

It often happens that, because vibrations are combined with other professional dangers and harms, we cannot establish a clear causal relationship between the vibration effects and health damages. However, numerous studies and researches show that vibrations have an effect on human health and that a shorter, but constant, exposure to high vibration values can cause pain in the stomach and chest, shortness of breath, nausea and dizziness, whereas a long-term and constant exposure can lead to workers' psychomotor, physiological and psychological system disorders. The majority of drivers are generally not aware of the harmful effect of vibrations mostly because of the simultaneous activity and effect of many negative factors [2,3].

The harmful effect of vibrations is particularly emphasised in older tractors, where there is no effective system of vibration and road-bump amortisation [4]. The largest number of tractors functioning in middle-income countries is precisely without quality suspension systems. Therefore, in these tractors, it is necessary to develop mechanisms to reduce vibrations and protect the drivers from the harmful influences on health. In older tractor models, the installation of a quality suspension system, on the cabin or on the axles, is a technical solution with uncertain success and large financial investment. In such cases, the better approach is to upgrade the seat, which not only increases the driver's comfort but it also decreases the influence of vibrations on his/her body. Some studies have indicated that some simpler solutions regarding the design of the seat can be more effective in the reduction of the level of exposure of the driver to vibrations than some much more complex solutions [5]. In addition, such solutions are attractive to many farmers because of their low price.

This paper considers the possibility of optimisation of oscillatory comfort of tractor seats by the addition of vibroisolation elements – cushions, on the existing seat. The appropriate parameters were monitored on the seats of IMT 539 and IMT 533, the most frequently used tractor models on the fields in Serbia, during ploughing operations, as well as on Ursus C-355 tractor during the transport of materials in the basket. The ability of different fluids and materials to isolate vibrations and prevent them from spreading to the driver, and therefore ensure the driver's comfort, was also investigated. Having in mind the possible alternatives (a cushion individually filled with air, water, hard-pressed sponge) and based on the set criteria (comfort, price, reduction of vibrations), the multi-criteria analysis was performed in order to select the most optimal solution, whereby the reduction of vibrations was the primary criterion. The analytic hierarchy process (AHP) method was used in the analysis, and the best solution according to the set criteria was obtained by using the software Expert Choice 11.

2. CRITERIA AND ALTERNATIVES

The analytic hierarchy process (AHP) method is one of the most famous and, in recent years, the most frequently used methods for multi-criteria decision-making. This is a multi-criteria technique based on the decomposition of a complex problem into a hierarchy in which the goal is placed on the top of the hierarchy, whereas criteria, sub-criteria and alternatives are positioned on the lower levels. This method compares the advantages and disadvantages of certain alternatives and, as the final result, gives the priorities of alternatives in the form of a number [9].

The analysis for the optimisation of oscillatory comfort of tractor seats was carried out based on the criteria for three alternatives – cushions with different fillings. The following alternatives were examined (Image 1):

1. a cushion filled with air
2. a cushion filled with water
3. a cushion made of hard-pressed sponge

hard-pressed sponge *air* *water*

Image 1 *Alternatives*

The criteria were chosen regarding their importance when working with a tractor, and they were: general comfort when driving, ability to reduce vibrations and cushion price (Image 2). During the optimisation, some of these criteria should be maximised (comfort and reduction), and some of them should be minimised (price). The ranking of the importance of a criterion should be determined according to the importance factor of the criterion, depending on the initial task set.

In this paper, the cushion's ability to reduce vibration was chosen to be the most important criterion, having in mind that the primary objective was the increase of the seat's oscillatory comfort. The same procedure of the analysis can also be conducted in case of appointing higher importance to some of the remaining two criteria or in case of appointing equal importance to all three criteria.

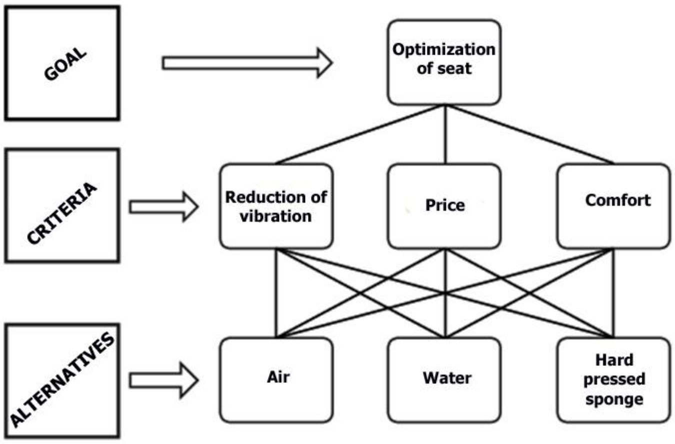


Image 2 *Specific hierarchy elements in multi-criteria analysis*

For the purposes of this analysis, the intensity of vibration acceleration on the seats of IMT 533, IMT 539 and Ursus C-355 tractors was measured (Image 3) [6,7]. These types of tractors prevail both in Republic of Serbia and in most of underdeveloped countries.

(a) IMT 533 *(b) IMT 539* *(c) Ursus C-355*

Image 3 *Tractors used for the measurements*

All tractors has simple mechanical seats, where IMT 539 had the seat upholstered with sponge, and IMT 533 and Ursus tractors had bare metal seats, on which the drivers usually put improvised cushions for better comfort (Image 4).

(a) IMT 533 *(b) IMT 539* *(c) Ursus C-355*

Image 4 *Tractor seats used for the measurements*

3. ANALYSIS

Reduction of vibrations: A seat's ability to reduce vibrations is verified based on the measurement of acceleration of vibrations on the seat and their evaluation in order to get the daily levels of driver's exposure to vibrations. The reduction of vibrations is a quantitative criterion because there are concrete values which indicate the cushion's quality regarding this criterion. The measurement process was performed in accordance with ISO 2631, and Brüel & Kjær 4447 device was used.

If we analyse maximum RMS values of vibration acceleration and daily levels of driver's exposure to vibrations for each of the cushions individually, it can be noticed that the levels of exposure are reduced several times in comparison to the original seat (Table 1 and Table 2).

Table 1. *Maximum values of vibration acceleration*

Tractor model - operation	Cushion type			
	Original seat	hard pressed sponge	air	water
	maximum RMS value [m/s^2] (axis)			
IMT 539 - plowing	8,942 (X)	2,494 (X)	2,083 (X)	2,803 (X)
IMT 533 - plowing	5,780 (Z)	1,273 (X)	0,824 (X)	0,924 (X)
Ursus C-355 load transport	7,761 (Z)	6,079 (Z)	4,707 (Z)	4,198 (Z)

Table 2. *Levels of daily exposure to vibrations*

Tractor model - operation	Cushion type			
	Original seat	hard pressed sponge	air	water
	Level of daily exposure A(8) [m/s^2] (critical axis)			
IMT 539 - plowing	12,52 (X)	3,49(X)	2,92 (X)	3,92 (X)
IMT 533 - plowing	5,780 (Z)	1,78 (X)	1,15 (X)	1,29 (X)
Ursus C-355 load transport	9,76 (Z)	6,26 (X)	4,707 (Z)	4,64 (X)

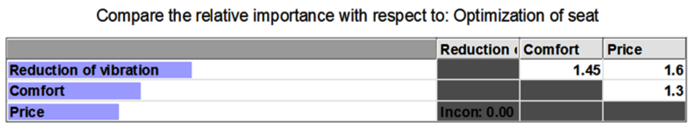
By comparing these values with legally permitted limits in professional conditions, it can be seen that only the cushion filled with air was able, in one case, to bring the daily level to the borderline value of 1.15m/s2 [8]. All values that exceed the limit show that the drivers should perform their activities for less than 8 hours, which is the usual time of one shift. For these reasons, it can be concluded that all cushions showed satisfactory results regarding the reduction of vibrations on the tractor seat, and that the cushion filled with air should, as an alternative, have the advantage in relation to the others, according to this criterion.

The price of the cushion: The price of the cushion also represents a quantitative and objective data, with a note that the prices of factory-produced cushions vary depending on the manufacturer and quality. A higher price of cushions filled with air and water contributed to their lower mark in comparison to the cushion made of hard-pressed sponge.

Comfort: This is a qualitative criterion because the evaluation is performed based on the subjective assessment of the driver. The driver's feeling of general comfort, as a criterion, was evaluated with the highest marks in case of the cushion filled with air. In case of the cushion filled with water, the subjective feeling of comfort was not optimal because the drivers reported temperature discomfort, that is, the cushions were too cold. The measurements were conducted in the winter and spring, at the outdoor temperature between 5 and 15°C, and since the tractors did not have cabins, the water in cushions quickly gained the outdoor temperature and it represented a discomfort to the drivers. Although in a tractor with a cabin and in better temperature conditions the feeling of comfort would probably be better, the research is related to the least favourable, and not the most favourable working conditions, so this criterion was evaluated with low marks for cushions filled with water.

4. RESULTS AND DISCUSSION

When comparing the criterion 'reduction of vibrations' to 'comfort', the first criterion has a moderately higher importance in comparison to the second (1.45), and 'comfort' has also a moderately higher importance in comparison to the criterion 'price' (1.3). The criterion 'reduction of vibration' is more important than 'price' (1.6) (Image 5).


Compare the relative importance with respect to: Optimization of seat

	Reduction (	Comfort	Price
Reduction of vibration		1.45	1.6
Comfort			1.3
Price	Incon: 0.00		

Image 5 *Determination of the importance of the criteria by comparing them in the program Expert Choice*

The software performs the normalisation and calculates the mean values for each criterion. In that way we obtained the criterion importance values of the objective 'seat optimisation'. The most dominant criterion was 'reduction of vibrations' (0.431), and the least significant criterion was the 'price' (0.255) (Image 6).

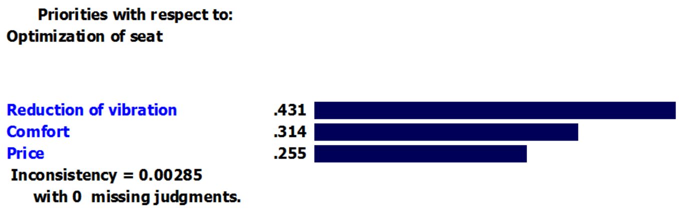


Image 6 *Criterion importance values by priorities*

The inconsistency of the decision-maker in the process of criteria comparison and evaluation was 0.00105, which is considered to be more than acceptable (<0.10).

The next step was the determination of the relevant importance of the alternatives in relation to the given criterion. When we compare the alternatives according to the criterion 'reduction of vibrations', the alternative 'air' is better than 'water' (1.9) and 'hard-pressed sponge' (2.1), whereas 'water' is slightly better than 'hard-pressed sponge' (1.2) (Image 7).

According to the criterion 'comfort', the alternative 'air' is better both than 'water' (2.1) and than 'hard-pressed sponge' (1.75), whereas 'hard-pressed sponge' is slightly better than 'water' (1.35)

According to the final criterion, 'price', the best quality has 'hard-pressed sponge' in comparison to 'water' 2.1 and with 'air' 2.3, whereas 'water' is slightly better than 'air' (1.1).

The final values of the priority of alternatives are presented in Image 10, according to which the most favourable alternative is ‘air’ with the total priority of 0.429, and the least favourable alternative is ‘water’ with 0.25 (image 8).

Compare the relative importance with respect to: Reduction of vibration

	Water	Hard press	Air
Water		1.2	(1.9)
Hard pressed sponge			(2.1)
Air	Incon: 0.00		

Compare the relative importance with respect to: Comfort

	Water	Hard press	Air
Water		(1.35)	(2.1)
Hard pressed sponge			(1.75)
Air	Incon: 0.00		

Compare the relative importance with respect to: Price

	Water	Hard press	Air
Water		(2.1)	1.1
Hard pressed sponge			2.3
Air	Incon: 0.00		

Image 7 *The quality of alternatives according to the criterion ‘reduction of vibrations’*

Image 8 *Final values of the alternatives*

In case of the change of the input data, the stability of the obtained solution is determined with the sensitivity analysis. The software allows this kind of analysis in several ways. One of them is the application of the performance sensitivity diagram. Image 9 presents the influence of the criteria on each alternative for a particular case.

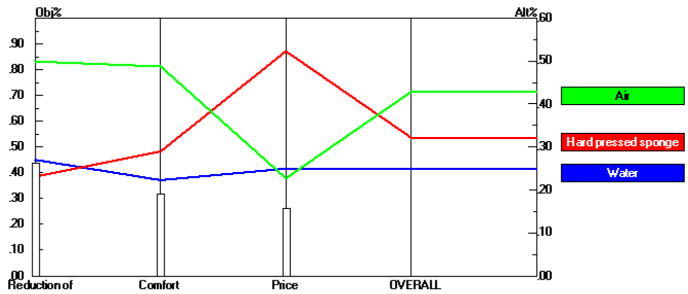


Image 9 *The sensitivity performance diagram according to the set importance of the criteria*

In Expert Choice, this diagram is dynamic since it allows the withdrawal of the criteria columns and modification of their importance values on the y-axis of the diagram. As a response to these changes of importance values, there are changes in the value of the priority of alternatives in the assisting (right) y-axis of the diagram.

If we would like to reconcile all set requests, that is, if equal importance was given to all criteria, the best or the most optimal alternative would still be the air-filled cushion (0.426), whereas the next alternative would be cushion filled with hard-pressed sponge (0.348). The worst alternative, in this case, would be the cushion filled with water (0.227).

5. CONCLUSION

As a result of the conducted multi-criteria analysis with the AHP method, the optimal solution by the set criteria was obtained. The highest importance was given to the criterion ‘ability of the cushion to reduce vibrations’ because of the desire to achieve the improvement of the oscillatory comfort of the tractor. In such conditions, as the optimal solution, we obtained the alternative – the cushion filled with air.

The multi-criteria analysis allows us to also get the optimal solution with other relations of criteria importance, depending on the previously set task. Therefore, the software Expert Choice, which performs all mathematical operations in the background without burdening the user, provides great assistance

ACKNOWLEDGEMENT

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REFERENCES

[1] A.J.Scarlett, J.S.Price, R.M.Stayner, (2007) Whole – body vibration: Evaluation of emission and exposure levels arising from agricultural tractors, Journal of terramechanics, 44, pp. 65-73.

[2] H.C.Boshuizen, et al., (1990) Self-Reported Back Pain in Tractor Drivers Exposed to Whole-Body Vibration, International Archives of Occupational and Environmental Health, 62, pp. 109-115.

[3] B.Cvetanović, J. Jovanović , (2013) A review of harmful effects to the health of tractor drivers from the impact of whole body vibration, Tractors and power machines, 18(3), pp. 58-65.

[4] B. Cvetanović, B, M.Cvetković, D.Cvetković, M.Ristić, (2014) Effect of the Suspension on Whole Body Vibration: Comparison of High Power Agricultural Tractors, VIII International conference Heavy machinery-HM 2014, Serbia, G7-11

[5] N.Ozkaya, D.Goldheyder, B.Willems, (1996) Effect of operator seat design on vibration exposure, American Industrial Hygiene Association Journal, 57, pp.837–842.

[6] Cvetanovic B. and Zlatkovic D., Evaluation of whole-body vibration risk in agricultural tractor drivers. Bulgarian Journal of Agriculture Science (2013),19, pp. 1161-1166

[7] Cvetanović B., Zlatković D., Cvetković D., Janjić N., Janjić Z., Whole body vibrations in older IMT tractor models. Journal of Balkan tribological association (2015), 21 (4), pp.859-864

[8] International Organization for Standardization (1997) ISO2631-1:1997 Mechanical vibration and shock – Evaluation of human exposure to whole-body vibration – Part 1: General requirements.

[9] B.Agarski, (2014) Razvoj sistema za inteligentnu višekriterijumsku procenu opterećenja životne

[10]Sredine kod ocenjivanja životnog ciklusa proizvoda i procesa, Doktorska disertacija, Univerzitet u Novom sadu, Fakultet tehničkih nauka.

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MAINTENANCE OF WORK EQUIPMENT BASED ON VIBRATION DIAGNOSIS FOR THE PURPOSE OF EMPLOYEES' SAFETY

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Abstract - *Maintenance of work tools and equipment is one of the major measures that can significantly affect the safety and health of employees. However, if we do not pay enough attention to this problem, the consequences could be improper functioning of the equipment, which, due to failure can cause the breakdown and employees' injuries. For this reason, the paper pays special attention to preventive maintenance and failure analysis of vital parts, equipment and tools. In addition to the concept of preventive maintenance, which is given importance in the work, this poper focuses on the concept of equipment maintenance based on reliability. Several examples of improper maintenance of equipment and failures that can lead to serious consequences for both employees and have been presented here.*

1. INTRODUCTION

Maintenance of work tools and equipment involves implementation of technical and administrative activities in order to complete a certain function without failure for a specified period of operation. In other words, maintenance of the equipment represents a "combination of technical and administrative activities during the life cycle of equipment for work that needs to keep its state in which it can perform its function," [1]. Apart from these wide consequences, maintenance of work equipment plays an important role that influeces the safety and health of not only the employees who are directly involved in the maintenance, but also other employees in the company. Occupational injuries can occur during the maintenance process in various cases, for example: employees who perform maintenance jobs on machines can be injured if the machine is accidentally started, they are exposed to hazardous substances or if they have to take a non-physiological posture of their body during work. Poor quality of maintenance may also contribute to serious consequences (eg. if inappropriate parts for replacement or repair of machinery are used). Failure to perform maintenance shortens the life of equipment and facilities which may cause accidents and threaten the environment Maintenance has two very important roles. Firstly, regular maintenance, which is properly planned and performed is very important since it ensures safe and reliable operations. Secondly, the maintenance of the should be safely carried out, provided with adequate protection for maintenance staff as well as all persons who find themselves in the space.

2. CONCEPTS OF EQUIPMENT MAINTENANCE

Historically, the concept of the maintenance system has been developed from the simplest - corrective approach to restore the system from the state of the termination of the state of the work, to the most complex - the concept of self-preservation. Only the concept of maintaining work equipment is shown in Figure 1.

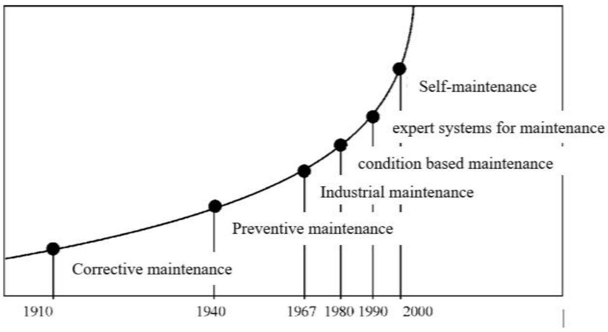

Fig. 1 *The development of the concept of maintenance*

As seen from the Figure 1, the concept of equipment maintenance has been changing throughout time. At the beginning of the last century, the maintenance was mainly enforced in case of equipment failure. The equipment was usually repaired for its reuse. Only in the middle of the last century did preventive maintenance, repairs and replacement of parts before the fault (dismissal) become significant. The rapid development of industry in the sixties raised the concept of industrial maintenance. Considering the problems in this area in the eighties, the maintenance concept referred to the state in which the equipment was able to perform its intended function. Development of science, especially in the nineties, contributed to the concept of expert system maintenance. At the beginning of the twenty-first century, the maintenance was given a completely different concept which was focused on self-maintenance. The future of maintenance is difficult to predict, due to new develoments and rapid replacement of obsolete equipment.

The concept of self-maintenance is related to the "robotic" way of production. The reason for this attitude lies in the fact that all new technical systems and equipment in addition to basic functions, have functions whose purpose is self-diagnostics of embedded parts. Secondly, an important prerequisite for the realization of self-maintenance concept is new technology that makes space for diagnosis and self-repair – more exactly cloning machine parts and performing work without the presence of workers ("factories without people").

During its lifetime, the equipment is exposed to broad spectrum of external and internal disturbances, leading to discrepancies in their basic characteristics and parameters of the state from its nominal (permissible) values. In the ongoing pursuit of users to maintain the system performance within the permissible deviations or return it to the (desired) interval, various systems of maintenance have been developed depending on the characteristics of applied concept, technology and maintenance organization [2].

Maintenance of technical system includes procedures, methods and techniques that ensure the correct operation of the system in a certain period of time, in order to prevent or delay the system failure (preventive maintenance) and (repair (corrective maintenance).

Choosing the concept of preventive maintenance and applying the appropriate models in technical systems can contribute to positive effects in terms of increasing safety and reducing maintenance costs. In the initial development of the industrial systems, maintenance was considered a secondary activity, and the impact of maintenance on production efficiency and increased productivity was considered insignificant. Nowadays, however, maintainance of the equipment occupies an important place in the production system. This is explained by the fact that the slow growth of industrial production, together with the rising price of raw material and reduction of capacity, may be slightly improved by efficient maintenance of work equipment, which is an integral feature of the production process. The starting point is the fact that the maintenance is a part of total production, and that it is defined as the permanent control of the work tools which, if necessary, should be subjected to certain repairs to keep up with a continuous production process and increase the safety of employees.

Among all activities that accompany the production, maintenance has been receiving increasing importance and there is no doubt that it has aroused interest due to its ability to significantly extend the lifetime of machinery and equipment in all sectors of production. Later, the vision of the maintenance process changed with the introduction of mechanization and automation and increased number of machines being used. Development of industry brough about an organized manner of maintenance that have a preventive role in the function of security.

Equipment maintenance involves primarily the concept of maintenance, which essentially represents the moment in which the decision to maintain is being made: before failure (preventive maintenance), after the failure (corrective/ recovery maintenance) or combined. Technological aspects of maintenance is reflected in the type and method of performing maintenance procedures. One can say that the maintenance includes procedures, methods and techniques that ensure the correct operation of technical systems in a given period of time, in order to prevent cancellation. There are several concepts of maintenance, among which the following two concepts are prevailing (corrective and preventive)

Corrective maintenance means that the activities are aimed at the renewal of work equipment when it fails to function properly (eg. the repair or replacement of damaged parts). This concept of maintenance is also known as "reactive maintenance" because the maintenance activity begins when an unexpected termination (malfunction) happens. Correction is made after the failure so that the equipment could be functional and used in a safe manner. This is an unscheduled maintenance concept that usually brings greater danger and risk to employees in relation to the concept of preventive maintenance.

Preventive maintenance implies that activities are carried out at predetermined time intervals or according to prescribed criteria to minimize possible untimely failure (fault). In this case, the maintenance activities are pre-planned, proactive and focused on process control monitoring equipment (eg. replacement, lubrication, cleaning and checks).

Monitoring and diagnostics by means of vibration

Monitoring the condition of machinery and equipment is only one of the activities that are part of technical maintenance for whose purposes we should measure and analyze vibrations. In principle, the use of vibration is represented in the implementation of the seven main groups of activities within the technical maintenance of mechanical systems, those being:

1. Monitoring the condition of machines and equipmen
2. Diagnostics of machines and equipment
3. The balancing of the impeller with vibration signal
4. Detection of the sources of vibration
5. Detection of acoustic emission sources
6. Modal analysis
7. The ultrasonic detection of damage.

Each group of activity has certain limitations, both in terms of achieving the goals and tasks, as well as in possibility to display the basic characteristics of vibration in order to achieve optimal results. For the presentation of the results, it is necessary to separate the symptoms of vibration frequency and manner of their occurrence. In terms of frequency, there are four bands or ranges: the low frequency range, the range of medium frequency, the range of high frequency and the range of ultrasonic frequency. According to the mode of occurrence, vibrations could be of natural origin (caused by dynamic processes in machines) and artificial vibrations triggered by special sources - oscillators. Natural vibrations are used to study the problem and solving problems within the first five groups of the above mentioned activities in technical maintenance of machines, while the artificial vibrations are applied in the remaining two cases.

3. MONITORING OF MACHINE CONDITION BY VIBRATIONS

Machine condition monitoring is the process of monitoring the condition of a machine with the intent to predict mechanical wear and failure. The objects of monitoring machine condition in terms of its vibrations are machinery and equipment that are sources of vibrations. These machines have vibration forces caused by e.g. movement of the individual elements, flow of fluid, or as a result of the operation of an alternative magnetic field. In rare cases, the monitoring object may be equipment that is not itself a source of vibration force and vibration, and whose operation is influenced by other sources of vibrations from the environment.

The aim of machine condition monitoring by vibration is to detect changes in vibrations present in the object that is being tested. The causes of such changes are most commonly consequences of certain defects. Machine condition monitoring is carried out first vibration test at low and medium frequencies that are very well spread from their place of origin to the place where their control is carried out. The number of such places near each machine can be reduced to one or two if the machines have a common housing. Vibration measurements can be performed without any changes to the operation of the facility. If this is not contained in the system of immediate protection, the monitoring system can be fitted with (instrument) a measuring wheel through which all connected inverters used in the monitoring system. This procedure dramatically reduces the price of the monitoring system, while not reducing the reliability of the results obtained.

The task of the system for monitoring the condition is to:

- measure the sizes important for reliable operation of individual plants,
- analyze measurement signals, or calculate certain parameters that allow direct assessment of the situation of each facility / equipment, and
- alarm, or trigger the operating part of the plant (protections, controls) in the case of impermissible level or the mutual relations of controlled size.

3.1 Systems for continuous monitoring of the condition

Systems for continuous (ongoing, continuous, on-line) monitoring are commonly installed on new installations or already existing installations.

One of the areas with intensive development of a system for monitoring the condition is the area of monitoring mechanical quantities, especially vibrations. The leader is the development of a system for monitoring the status of power plants (turbo and hydro aggregates, pumps, fans and a large fig.). These systems can be divided into:

- conventional systems for monitoring and
- systems for diagnostic monitoring.

Schematic representation of the components of a complete system for monitoring the status is shown in Figure 2.

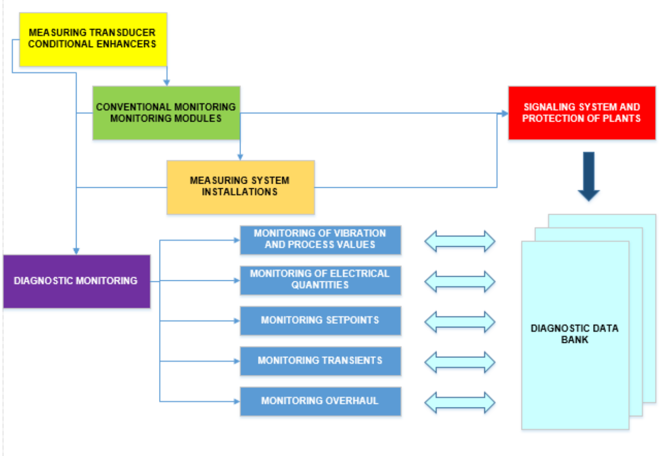

Fig.2 *Block diagram of a complete system for continuous condition monitoring*

3.2 Systems for controlling conventional monitoring

Conventional monitoring systems consist of:

- measuring transducer, which have the task of measuring and
- modules, which have the task of accepting and adjusting (conditioning) signal from the measuring encoder, their basic processing, comparing with the standards prescribed values, and alarm activation and execution protection.

Such systems are commonly saved as separate "hardware" of the unit.

Customize the system for conventional monitoring carried out according to current applicable standards. For monitoring the condition of rotating machinery vibration are currently using authoritative standards ISO 13373-1: 2002, which contains general procedures and ISO 13373-2: 2005, which contains the methods of processing, analysis and presentation of vibration data. The primary purpose of these systems is protection against excesses, or activate signaling switching and protection in the event of a monitored magnitude exceeds the value of this type of facility permitted to the appropriate standard.

4. THE IMPACT OF MAINTENANCE ON OCCUPATIONAL SAFETY AND HEALTH

Maintenance is carried out in all sectors and at every workplace. This is not the exclusive domain of technicians and engineers for maintenance, but maintenance is included in the daily tasks of most employees. Therefore, employees who work on maintaining exposed to a wide range of hazards and damage (mechanical, chemical, physical, biological or psychological). Employees who perform maintenance can be at risk of:

- occurrence of diseases of the musculoskeletal system working in adverse postures of the body, sometimes in difficult weather conditions (eg, cold);
- occurrence of respiratory problems that are related to exposure to asbestos (eg. Maintenance of old buildings or industrial facilities);
- occurrence of skin diseases due to contact with dangerous and harmful chemical substances (fats, oils, solvents, acids, etc.);
- diseases that are the result of exposure to biological hazards;
- various types of injuries, including falls from height, slipping, tripping, hitting the moving parts of equipment and machinery.

Thus the maintenance staff are exposed to the risks of developing various types of injuries. The possibility of injury from moving machine parts that can cause injury or death should be borne in mind at all times

Regardless of whether the maintaining activies are small or extensive, they could adversely affect the safety and health of not only the employees who perform these activities, but also other persons who are in the area. The injuries of maintenance staff can occur in different cases, namely: in the random startup equipment, exposure to ionizing radiation or hazardous substances. Poor maintenance can cause safety problems, for example in case of inappropriate parts for replacement or repair which can cause serious injury to employees and damage to the equipment.

The persons employed at maintenance are exposed to noise, vibration, radiation, hazardous substances, vapors and gases. However, despite these identification employees who work on maintaining the open space could mention the heat in summer, cold in winter and humidity. The data show that about 20% of all occupational injuries occur in connection with maintenance. Statistical figures from several European countries show that about 10-15% of all fatal injuries were related to maintenance, [3].

Scientific studies suggest that the diseases and health problems related to work (such as asbestosis, cancer, hearing problems and diseases of the muscular - skeletal system) are also most common among employees who deal with maintenance. For example, in the activities in the energy sector (services for supply of electricity, gas and water), the maintenance jobs have the largest number of injuries.

The question of when it is the right time to start maintenance is always significant. It is recommended that the maintenance process should begin in the design stage and planning before employees begin maintenance work process. It is essential that the application of appropriate risk assessment procedure for maintenance, as well as to take into account relevant preventive measures to ensure the safety and health of employees involved in maintenance. Since maintenance tasks are completed, special checks (visual inspection and trial testing the functioning of technical systems) should be carried out in order to safely be said that the maintenance is properly performed and that were not created new risks. Throughout the maintenance process, we should take into account whether the maintenance was carried out as planned and whether the equipment and jobs left in a safe condition for a continuation of the work process. More details and information about this are given in http://www.cen.eu, [5].

5. SAFE MAINTANCE

Safe maintenance is a process that begins before an actual work and ends after the verification and recording of activites and technical documentation. Participation of employees in all phases of maintenance increases, not only the safety of the process, but also the quality of work.

Hereinafter described are the rules for safe maintenance relating to:

a) Maintenance plan,
b) Work in safe conditions,
c) Use of appropriate equipment,
d) Compliance with safe work procedures,
e) Checking the quality of work performed.

a) Maintenance plan

As part of planned maintenance is necessary to make a risk assessment for all activities on maintenance of work equipment. In this activity it is necessary to involve all employees and consider the following:

- The volume of tasks that need to be performed, time required and how it will affect other employees and jobs in the workplace;
- Identify the hazards and harmful effects, for example. of electricity, exposure to hazardous substances, the presence of dust / asbestos in the air, limited space, moving machine parts, the fall from a height or through the floor, heavy objects to be moved, parts of which are inaccessible or difficult to reach;
- To determine what is required to perform maintenance (required expertise and the number of employees in maintenance, who will be involved, what role individual persons have, responsibility for contacts with employees of the contractor or the customer, management tasks, which inform about potential problems, tools to be used, means and equipment for personal protection at work, and other measures of health and safety at work which may be necessary);
- Secure access to the work zone and method for rapid evacuation in case of emergency;
- Enable all employees who work directly in maintenance and inform all those who work near them, the jobs (to ensure competence of employees and their safety), the responsibilities and all the procedures that will be used during the maintenance process, including reporting problems. This is especially important if the maintenance is performed by a subcontractor.

All employees of the maintenance should be included in the planning phase of maintenance because they are best placed to recognize the danger and harmfulness of the most effective ways to overcome them. Analysis of risk assessment and planning phase of the document to be delivered to employees who participate in the performance of maintenance, as well as other employees that maintenance can be affected. Involving employees together with subcontractors in training and familiarization with the established procedures are very important elements in ensuring their safety.

b) Work in safe conditions

Procedure as defined in the risk assessment should be applied in practice during maintenance. For example, make sure you must turn off the power source of energy of all technical systems which perform maintenance process and according to the agreed procedure. It is necessary to set up a visual warning label (date and time of exclusion and name of the person authorized to remove the ban placed an inscription). Employees should check whether there is a safe entrance to and exit from the working zone, in accordance with the plan of carrying out the works.

c) Use of proper equipment

Employees who perform maintenance tasks need to have the proper tools and protective equipment, which may differ from those commonly used. This is because the work in the maintenance may be performed in a space that is not an ordinary workplace, where employees are exposed to many dangers and hazards. Therefore, employees must have adequate means and equipment for personal protection at work. For example, employees who clean or replace filters on the exhaust fan may be exposed to dust concentrations that are much higher than normal in this workplace. Access to these filters, often located on the roofs, must also be safe. Tool required for operation and means and equipment for personal protection at work that have been identified in the planning and risk assessment must exist (along with instructions for use, if necessary) and must not be used.

d) Compliance with safe work procedures

Procedures for safe operation adopted in the planning phase must be applied. Work Plan and the agreed deadlines must be respected. The activities should not be performed faster than the standardized time and procedures which it intended, because each "shortcuts" in business can cost a lot and lead to employee injuries or property damage. It is essential that line managers and / or other experts consulted in the event that something unexpected happens from the planned procedures. It is very important to respect the expertise and competence with command responsibility for the maintenance, because if this is not respected can lead to very serious consequences.

e) Quality Inspection Works

Each job after completing the intervention maintenance equipment must be checked whether it is done well. The verification should be done to make sure that the task is completed, that is part of where the maintenance was carried out in a functional and safe condition and that all of the material that emerged during maintenance removed. When all check and states that the process works well is completed, then the warning can be removed and start with a normal working process. Finally conducted checks need to write a report, which will describe the work done and give remarks about the difficulties at work as well as recommendations for improvement. Ideally you should also talked about this at a meeting attended by all the employees who were involved in the maintenance process, as well as those who work near them, so that they can give their comments and suggestions relevant to the maintenance process equipment improved.

Given the relatively large number of hazards and harmful effects that influence the risk for employees working on maintenance, it is necessary to take into account the estimated risk management. Before beginning maintenance work is necessary to do a detailed assessment of the risks and to include all phases of the business and all the dangers and harmful effects. This is particularly important to do in small and medium-sized enterprises because of their huge losses in case of injuries during maintenance.

Competence of employees who perform maintenance, including medical examinations and training is vital to safety. Although employees often expert in several areas of their competence for maintenance must be specially emphasized and the description of their tasks. However, activities that do not perform regularly need to further train. This is because employees are often violations if they try to perform tasks that are not trained or who do not have experience. Employers must check whether employees have the expertise to perform the necessary tasks, whether they are informed of the hazards and safe work procedures and whether they know what to do in situations that go beyond their expertise.

Employees often work on maintaining the high-risk areas. For these reasons, these works require the use of work equipment which is normally not used in the workplace, including means and equipment for personal protection at work. Procurement procedures must exist to ensure that the necessary tools and means of personal protection equipment provided to employees at work in the maintenance of technical systems. For example, you may need to temporarily lighting should be carried out in the explosion protection and to use appropriate personal protective equipment (eg. Respiratory protection when cleaning the filters). The risk during maintenance should be reduced to a minimum or even eliminated through a well-constructed equipment for work, the existence of appropriate tools and instructions from the manufacturer.

6. CONCLUSION

It is crucial to consider the maintenance as a process, not just a task that must be done temporarily. The maintenance process begins with the planning phase, and it includes comprehensive risk assessment. It is recommended that employees who work on maintaining or their representatives should be involved in the planning maintenance process. Procedures for maintenance must be respected, but conditions for solving the unexpected problems should also be present. Once the maintenance is completed, the functioning operations must be checked to make sure that the equipment or machinery and equipment are safe to be reused. Also, t the waste material should be removed, the area cleaned and trial operation conducted in order to keep up with the normal functioning.

The situation that can have negative impact on the safety of the employees is the fact that maintenance is often carried out within deadlines - "under pressure" – in order to quickly re-start the production process or to complete the job before the planned deadline. This problem increases if employees use work resources without necessary protective system. Risk assessment is done to prevent the possible consequences, to ensure that maintenance is performed safely and that employees who are engaged in the ongoing production process are not endangered, and that the equipment can beused again safely. The Risk Assessment Act for maintenance employees jobs should be kept together with the documentation on equipment maintenance.

Key elements in terms of safety and health of the staff employeed at maintainance are: training and defining the competence of employees working on maintenance, and respecting the safe work based on risk assessment. After the maintenance jobs are completed, specific checks (examinations and tests) need to be performed to make sure that the maintenance was carried out properly and that the equipment or workplace are left in a safe condition for future operations.

In the end, we can conclude that employers who do not maintain their work tools and equioment, especially those parts that are related to safety (eg. safety systems and alarms), have greater risks that equipment failure will be catastrophic for all employees.

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REFERENCES

[1] International Information Center of Safety and Health at Work: www.ilo.org/cis

[2] The European Agency for Safety and Health at Work (Bilbao, Spain): www.europe.osha.int

[3] The European Foundation for the improvement of living and working environment (Dublin, Ireland) www.eurofound.ie

[4] http://www.cdc.gov/niosh/homepage.html (access 09.20.2016)

[5] http://osha.europa.eu/en/topics/maintenance (access 09.20.2016).

[6] http://www.cen.eu (access 09.20.2016)

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