Influence of hardness and damping factor of porous material upon generated drum sound from impact of metal disk and metal plate

Snejana Pleshkova, Aleksandar Vasilev Kirechev, Dejan Ćirić, Tzvetan Nedkov

***Abstract*—Drum sound is a common source of noise in the industries. Noise generated by impact of two rigid objects with low internal losses is characterized by a high peak value and short duration. When two bodies collide, their kinetic energy transforms to another form of energy such as heat, plastic deformation, fracture and sound. In order to reduce the drum sound, contact surface material is investigated. The effects of materials with different properties and thickens are investigated here using the measurements carried out in laboratory conditions. Theoretical analysis of the discussed topic is reviewed in this paper but the calculation are not presented.**

***Index Terms*—drum sound effect, damping, internal loss, Hertz contact theory**

I. Introduction

Characteristics of noise generated by impact of two rigid objects with low internal losses include a high peak value and short duration. The mechanical impact phenomenon is a common source of noise in the industries. This phenomenon can happen in assembling lines with metal parts that drops on metal plates. The efficient way to reduce the impact noise is to place material with high damping properties on the metal plate. But despite the damping properties of material there is another factor that influences the intensity of generated noise – hardness defined by the module of elasticity. Collecting the experimental study of phenomena is not easy task because of high speed of propagation of elastic wave in solid structure.

In this paper, we investigate how those two parameters of porous material placed on the metal plate influence the noise levels generated by impact of falling metal disk on the metal plate. In order to investigate this influence several measurements are performed in a small anechoic chamber where the same material with different thicknesses and different elastic modules are used.

To predict the outcome of the experimental analysis, the theoretical approach and numerical modeling can be used and it will be reviewed it this article. The approach gives the model to predict for impact noise radiated by collision of two rigid cylinders are also used. Mechanical impact noise in many cases can be assumed to be similar to noise radiated by collision of two cylinders. Studies of the impact of cylinders on the basis of the Hertz contact model and acoustic theory as well as the contact impact force and acoustic pressure had been done.[1-3]

II. Theoretical Model

A. Hertz's Elastic Theory of Contact

The theory relates the circular contact areas of a sphere with a plane to the elastic deformation properties of the materials. Hertz states that the normal contact force between two colliding elastic bodies is given by [1-6], [11], [12]:

$$F = Kd^n \tag{1}$$

Where F is the contact impact force, d is the depth of penetration at the contact point and K is the contact stiffness parameter of the two materials which is given by:

$$K = \frac{4}{3}\sqrt{r}E^* \tag{2}$$

Where E^* represents the composite modulus of collision spheres and r represents the composite of collision sphere.

The time dependence of the contact force for a collision beginning at the time t_0 can be approximated by a half-sine pulse

$$S(t;t_0,\tau) = \begin{cases} \sin\left(\dfrac{\pi(t-t_0)}{\tau}\right) & \text{if } t_0 \le t \le t_0+\tau \\ 0 & \text{otherwise} \end{cases} \tag{3}$$

Where t is time, and τ is a time scale defined by:

$$\tau = 2.87\left(\frac{m^2}{rE^{*2}\mathrm{V}}\right)^{1/5} \tag{4}$$

V is the normal impact speed and the other constants used in (4) are defined as follows:

$$\frac{1}{r} = \frac{1}{r_1}+\frac{1}{r_2},\ \frac{1}{m} = \frac{1}{m_1}+\frac{1}{m_2},\ \frac{1}{E^*} = \frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2} \tag{5}$$

Aleksandar Kirechev is with the Faculty of Telecommunications, Technical University of Sofia, 8 Bulevar Kl. Ohridski, 1000 Sofia, Bulgaria (e-mail: sakata6371@gmail.com).

Snejana Pleshkova is with the Faculty of Telecommunications, Technical University of Sofia, 8 Bulevar Kl. Ohridski, 1000 Sofia, Bulgaria (e-mail: snegpl@tu-sofia.bg).

Dejan Ćirić is with Faculty of Electrical Engineering, University of Niš, Aleksandra Medvedeva 14, 18000 Niš, Seerbia (e-mail: dejan.ciric@elfak.ni.ac.rs).

r_i, m_i, ν_i and E_i are the radius, mass, Poisson ratio and Young's modules for spheres $i = 1, 2$. Force profiles of the form (4) have been used to model continuous contact forces for acceleration noise due to simple geometries.

B. Acoustic Analysis

The acoustic equation for the cylinder in a cylindrical coordinate system is given by [7-10]:

$$\begin{aligned}&\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial \Phi}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 \Phi}{\partial \theta^2}+\frac{\partial^2 \Phi}{\partial z^2}=\frac{1}{c^2}\frac{\partial^2 \Phi}{\partial t^2}\\ &p=\rho\frac{\partial \Phi}{\partial t}\\ &u_r=-\frac{\partial \Phi}{\partial r}\end{aligned} \tag{6}$$

where Φ is the velocity potential, c is the sound speed in the air, p is the sound pressure and u_r is the particle velocity.

If the cylinder vibrates with a velocity of $v_0 e^{j\omega t}$, the velocity potential $\Phi(r,\theta,t)$ becomes:

$$\Phi(r,\theta,t) = A_1 \cos\theta H_1^{(2)}(kr)e^{j\omega t} \tag{7}$$

where $H_1^{(2)}(kr)$ is the first order Hankel function of the second kind, ω is the angular frequency of oscillation and k is equal to ω / c. If plane of vibration is taken as the reference plane, the angle between the velocity of the cylinder surface and the plane of vibration is θ, then taking into account the boundary condition it can be written:

$$(u_r)_{r=R} = -\left.\frac{\partial \Phi}{\partial r}\right|_{r=R} = V_0 e^{j\omega t}\cos\theta \tag{8}$$

where R is the radius of the cylinder.

The velocity potential for any arbitrary velocity of the oscillating cylinder can be obtained by the Fourier synthesis of this solution when the Fourier transform of the velocity is known.

The sound pressure caused by the impact acceleration of a cylinder is given by:

$$p(r,\theta,t) = \rho\frac{\partial \Phi(r,\theta,t)}{\partial t} \tag{9}$$

III. Experimental Part

For the experimental part, the metal plate with thickness of 0.8 mm and dimensions 300 mm width and 400 mm length is chosen. The metal "disk" that hits the metal plate has a cylindrical shape with dimensions 30 mm radius and 100 mm height. The weight of the "disk" is 800 g.

The experiment is carried out in a small anechoic chamber at the Laboratory of Acoustics of the Faculty of Electronic Engineering, University of Niš, Serbia, see Fig. 1. Measurement setup consists of the microphone (Bruel&Kjaer Type 4144), pre-amplifier (Bruel&Kjaer Type 2619), power supply, sound card and laptop with recording software.

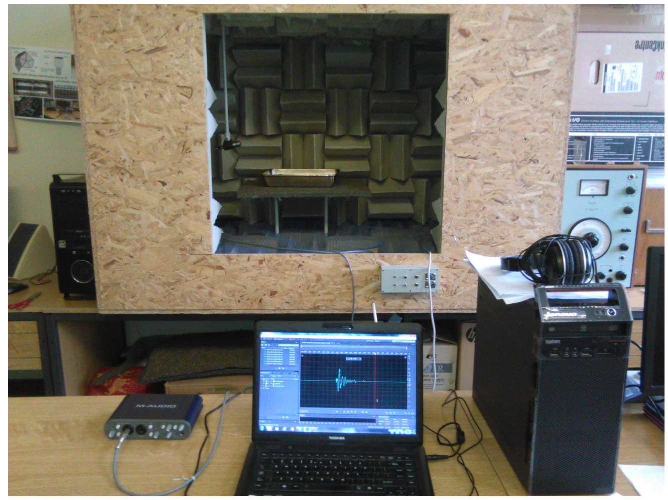

Fig. 1. Setup for the measurement of noise levels and characteristics in a small anechoic chamber at the Laboratory of Acoustics of the Faculty of Electronic Engineering, University of Niš, Serbia

In order to generate the impact noise, the metal cylinder is dropped from the high of 200 mm. In order to reduce this impact noise, the material with high damping properties is chosen. In order to investigate the influence of different hardness, two materials with different thinness and elastic modules are applied. The mechanical properties of materials are shown in Table 1. The materials' names are chosen by the hardness as "soft and "hard".

TABLE I
MATERIAL PROPERTIES

Property	Soft material	Hard material
Thickness	3 mm	6 mm
Compression strength	0,03 N/mm^2	0,10 N/mm^2
Module of elasticity	0,51 N/mm^2	1,20 N/mm^2
Compression hardness	0,015 N/mm^2	0,01 N/mm^2
Damping factor	0,48	0,47
Coefficient of friction (steel)	>0,50	>0,50

The experiment is performed with five different combinations:

- Only hard material placed on the plate;
- Only soft material placed on the plate;
- Combination of hard material and soft material (hard material is on top) placed on the plate;
- Combination of soft material and hard material (soft

material is on top) placed on the plate, and

- Combination of two hard materials placed on the plate.

For all of these combinations, the materials are not glued to the plate. To avoid mistakes, the measurement is repeated ten times for every combination.

IV. Results

The measurement results related to the spectrum analysis are shown in Fig. 2 Since absolute calibration of the sound pressure levels is not done, the level is given as relative ones.

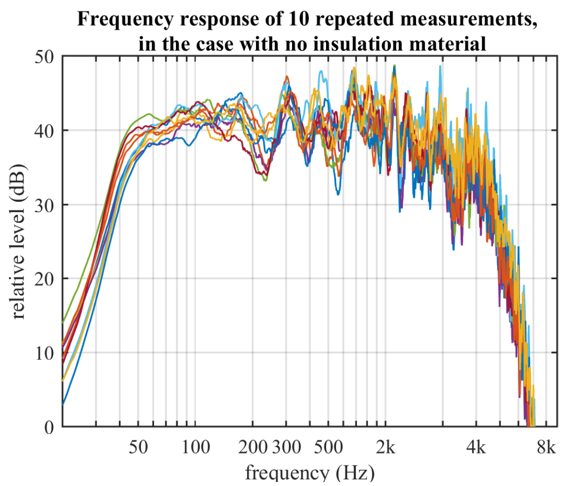

Fig. 2a. Spectra of impact noise generated when the metal disk hits the metal plate without damping material placed on top of the plate (noise levels are given in relative units, that is, dB)

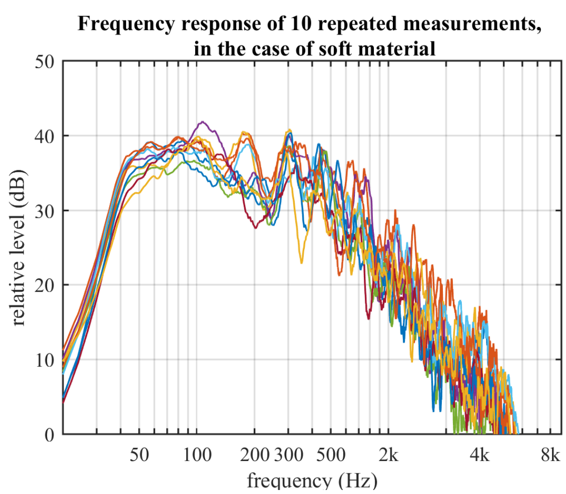

Fig. 2b. Spectra of impact noise generated when the metal disk hits the metal plate with damping material (soft) placed on top of the plate (noise levels are given in relative units, that is, dB)

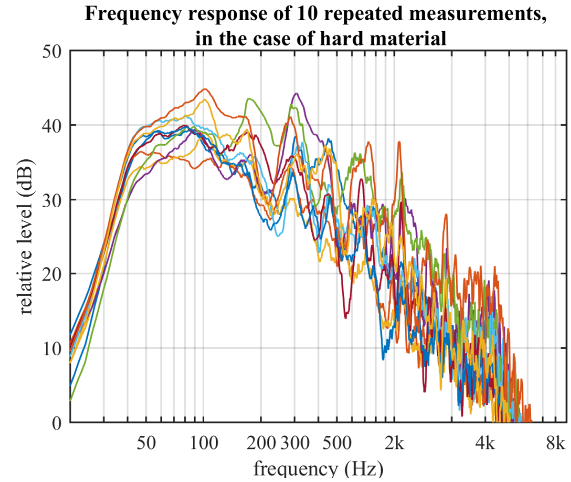

Fig. 2c. Spectra of impact noise generated when the metal disk hits the metal plate with damping material (hard) placed on top of the plate (noise levels are given in relative units, that is, dB)

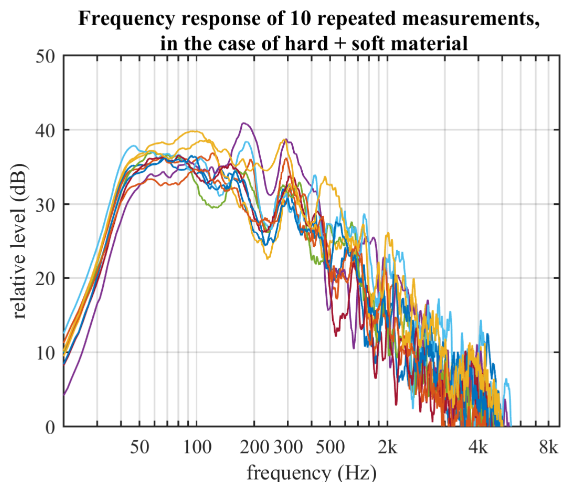

Fig. 2d. Spectra of impact noise generated when the metal disk hits the metal plate with damping material (hard + soft) placed on top of the plate (noise levels are given in relative units, that is, dB)

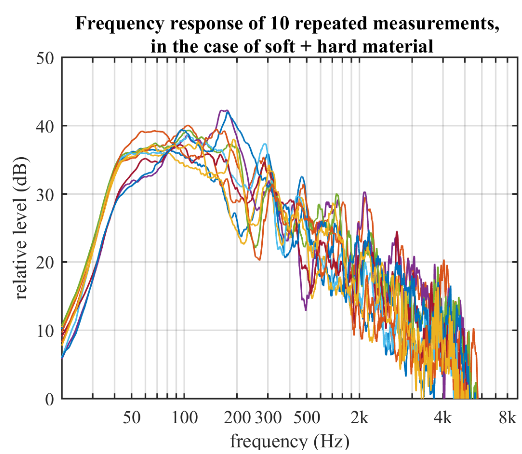

Fig. 2c. Spectra of impact noise generated when the metal disk hits the metal plate with damping material (soft + hard) placed on top of the plate (noise levels are given in relative units, that is, dB)

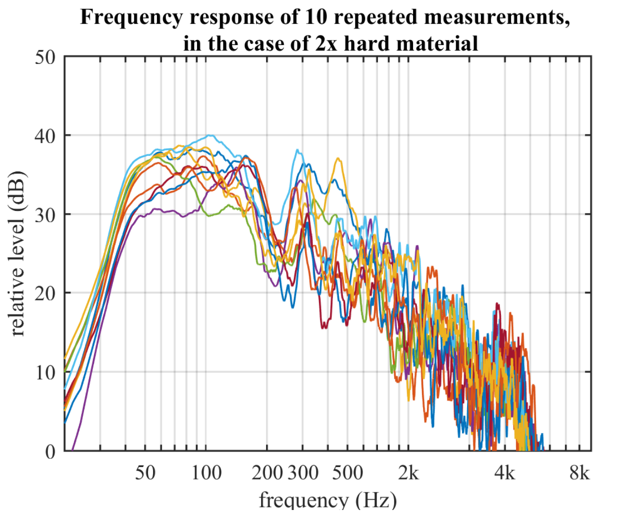

Fig. 2e. Spectra of impact noise generated when the metal disk hits the metal plate with damping material (hard + hard) placed on top of the plate (noise levels are given in relative units, that is, dB)

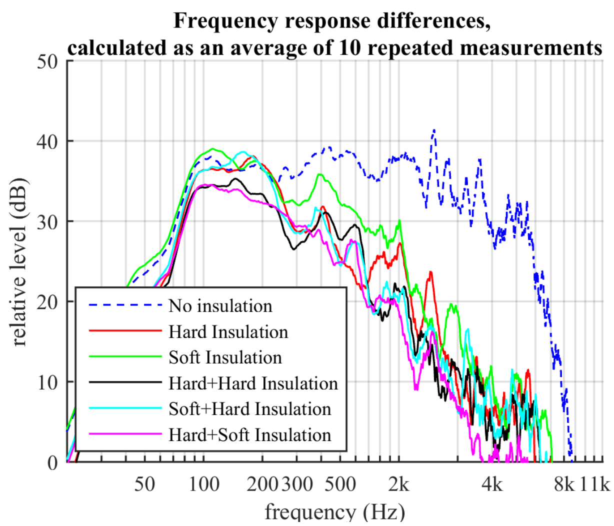

Fig. 3. Spectra of impact noise generated when the metal disk hits the metal plate without and with damping material placed on top of the plate (noise levels are given in relative units, that is, dB)

The results show that in the frequency range between 50 Hz and 200 Hz there is no reduction of noise level (in comparison to the case where no damping material is placed on the metal plate) when only one layer of damping material is applied. Application of double layer where the hard material is placed directly on the plate gives a reduction of only few dB in that frequency range. However, above 200 Hz the noise attenuation increases significantly for all combinations of materials. The noise attenuation at 300 Hz is between 5 dB to 10 dB, while it reaches the values of approximately 18 dB at 4 kHz.

Regarding the equivalent noise reduction (one-figure value), it can be calculated in different ways. Here, the following procedure is applied. The impact noises recorded without any damping material on the metal plate and with a particular damping material on the metal plate are first filtered with a filter of a particular pass-band. The ratio of overall sound pressure levels of these filtered noises represents the equivalent noise reduction for that pass-band, that is, frequency range of interest. If the frequency range of interest is set to be from 50 Hz to 11 kHz, the obtained results are presented in Fig. 3. The influence of low frequencies is rather significant since the noise levels at that frequencies are rather high. If the lowest frequencies (up to 100 Hz) are excluded from the calculation (by filtering out these frequencies), the equivalent noise reduction is increased as presented in Fig. 4 and 5. This case can be considered to be even more realistic especially from the psycho-acoustical point of view, since human ear is less sensitive to the lowest frequencies.

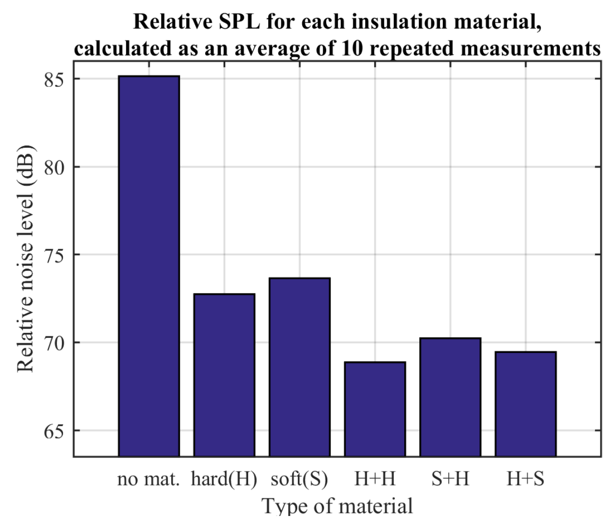

Fig. 4. Relative sound pressure (noise) level for different damping materials

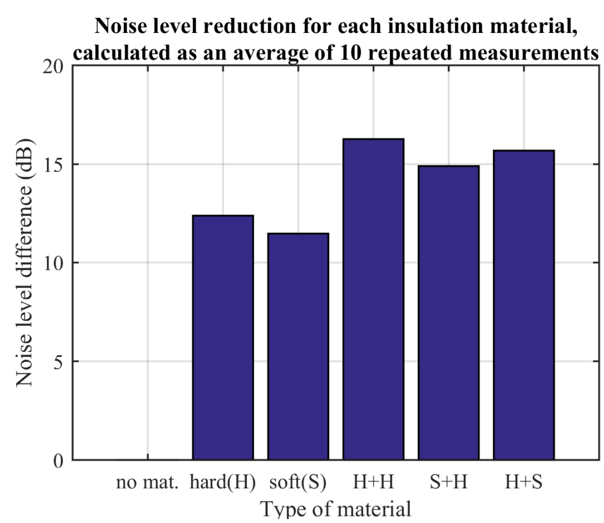

Fig. 5. Noise level reduction for different damping materials (taking into account frequency range from 100 Hz to 5 kHz)

V. Conclusion

The performance of different combinations of damping materials for reducing the impact noise strongly depends on the frequency range of interest. If we analyze the noise level reduction in a wide frequency range including the lowest frequencies from 50 Hz up to 11 kHz, we can see that the "soft" material reduces the overall noise level by 0.50 dB more than the "hard" material despite the lower thickness. But if we consider the frequency range from 100 Hz to 5 kHz, the noise reduction for the "soft" and "hard" material is almost the same. The situation is similar in the frequency range from 150 Hz to 5 kHz. However, the values of noise reduction are significantly greater for the latter frequency range, where they are about 11 dB and 12 dB in comparison to the former case where these levels are about 3 dB. Generally speaking, the best results are achieved using the double layer of the "hard" material. The greatest difference among the combinations of the damping materials with regards to the overall equivalent noise reduction exists in the widest frequency range of interest where the "hard" + "hard" material gives larger noise reduction by at least 1.5 dB in comparison to all other material combinations. However, when considering the performance of the impact noise reduction, the price and overall thickness should also be taken into account.

Acknowledgment

This work was supported by Technical University – Sofia inner program to research projects under 172PD0031-07: "Development of mathematical model to predict in studying the impact of various viscoelastic sound insulation materials on the sound insulation systems properties"

References

[1] H. Deresiewicz, A note on Hertz's theory of impact, *Acta Mechanica*, 6 (1) (1968) 110-112.

[2] K. E. Fallstrom, H. Gustavsson, N. E. Molin and A. O.Wahlin, Transient bending waves in plates studied by hologram interferometry, *Experimental Mechanics*, 29 (4) (1989) 378-387.

[3] K. Mehraby, H. Khademhosseini Beheshti and M. Poursina, "Impact noise radiated by collision of two spheres: Comparison between numerical simulations, experiments and analytical results" *Journal of Mechanical Science and Technology* 25 (7) (2011) 1675~1685

[4] Sn.Pleshkova, Al. Kirechev. Development of Method for Predicting the Influence of Elastic Porous Materials on Coincidence Frequency of Walls for Sound Insulation Based on Composite Structures. Comptes Rendus de l'Académie Bulgare des Sciences, Vol.70, No 1, pp.113-120, 2017

[5] Yinggang Li, Tianning Chen, Xiaopeng Wang, Kunpeng Yu and Chao Zhang, "Theoretical and numerical investigation on impact noise radiated by collision of two cylinders" *Journal of Mechanical Science and Technology* 28 (6) (2014) 2017~2024

[6] J.A. Greenwood, "*Adhesion of Elastic Spheres*",. Proc. R. Soc. Lond. A (1997) 453, 1277~1297

[7] J. Chadwick, C. Zheng, D. James "Precomputed Acceleration Noise for Improved Rigid-Body Sound" *ACM Transactions on Graphics (TOG) - Proceedings of ACM SIGGRAPH* Vol. 31 Issue 4, July 2012

[8] Chaigne, and Lambourg, "Time-domain simulation of damped impacted plates. i. theory and experiments". J*ournal of the Acoustical Society of America* 109, 4, (2001), 1422–1432

[9] S. Schedin, C. Lambourge and A. Chaigne, Transient sound fields from impacted plates: comparison between numerical simulations and experiments", *Journal of Sound and Vibration*, 221(3) (1999) 471-490.

[10] K. E. Fallstrom, H. Gustavsson, N. E. Molin and A. O. Wahlin, Transient bending waves in plates studied by hologram interferometry, *Experimental Mechanics*, 29 (4) (1989) 378-387.

[11] G. Nishimura and K. Takahashi, Impact sound by mutual collision of two steel balls, *Bulletin of the Japan Society of Precision Engineers*, 1 (1965) 47-51

[12] E. J. Richards, M. E. Westcott and R. K. Jeyapalan, On the prediction of impact noise I: Acceleration noise, *Journal of Sound and Vibration*, 62 (4) (1979) 547-575.