DOI: 10.21062/mft.2020.019 http://www.journalmt.com

Study of Factors Affecting Vibration Damping Properties of Multilayer Composite Structures

Martin Vašina, Lumír Hružík, Adam Bureček
Faculty of Mechanical Engineering, VSB-Technical University of Ostrava, 17. listopadu 15/2172, 708 33 Ostrava-Poruba. Czech Republic. E-mail: martin.vasina@vsb.cz, lumir.hruzik@vsb.cz, adam.burecek@vsb.cz.

At the present time, mechanical vibration is undesirable in many cases. Therefore it is necessary to minimize unwanted vibrations in any appropriate manner. This paper is focused on a study of factors influencing vibration damping properties that were investigated using multilayer composite structures. Frequency dependencies of the displacement transmissibility over a frequency range of 2–1500 Hz were determined by the method of forced oscillations. It was found in this study that the vibration damping properties of investigated multilayer structures are significantly influenced by number of material layers, excitation frequency of mechanical vibration, applied materials in multilayer structures, inertial mass, material thickness and density. It was also observed that a superior ability to damp mechanical vibration leads to a shift of the first resonance frequency peak position to lower excitation frequencies.

Keywords: Multilayer Structure, Harmonically Excited Vibration, Displacement Transmissibility, First Resonance Frequency, Stiffness.

1 Introduction

Mechanical vibration is given by any motion that repeats itself after an interval of time [1]. In generally, mechanical vibration can be divided in several ways, for example:

- free and forced vibration,
- damped and undamped vibration,
- linear and nonlinear vibration,
- deterministic and nondeterministic (or random) vibration.

The mechanical vibration is an undesirable phenomenon in many cases, e.g. in precision engineering, vehicles, home appliances and scientific experiments. It can have a negative influence on manufacturing effectiveness and quality, tool life, equipment failures, safety at work, noise propagation [2 - 5] etc. For these reasons, it is necessary to reduce the unwanted mechanical vibration in any appropriate manner. There are various ways to eliminate the undesirable mechanical vibration. It is possible to apply leaf or helical springs, suitable damping pads, hydraulic shock absorbers etc. In many cases, for example in a mounting of processing machines, it is also necessary to ensure system stiffness.

The aim of this paper is to investigate vibration damping properties of multilayer composite structures which were subjected to harmonic excitations. Typical multilayer structures consist of three layers. The outer layers are made of high strength materials, whereas the core is made of lightweight materials [6]. In the case of the lightweight materials in the multilayer structures, it is also important to take into account their permissible operating load ranges depending on the type of loading. For example, the load ranges of the Regufoam® vibration 810 plus sample are as follows: the maximum static bearing capacity of 0.85 MPa, the maximum dynamic load bearing capacity for intermitted loading of 1.2 MPa and the short term peak load of 7 MPa [7].

The schematic diagram of a multilayer structure, whereas the layer number $i = \langle 1, n \rangle$, is shown in Fig. 1. In this case, the mechanical vibration is propagated through the multilayer structure from the input side "I" to the output side "O". The harmonically excited vibration is characterized by the force (F_I) and the velocity (v_I) amplitudes on the input side of the multilayer structure. Similarly, the response of the harmonically excited multilayer structure on its output (i.e. free) side is characterized by the force (F_O) and the velocity (v_O) amplitudes.

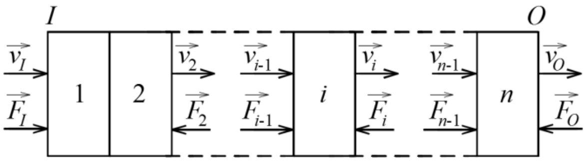

Fig. 1 *Schematic diagram of vibration propagation through a multilayer structure*

The schematic diagram of a damped system under the harmonically excited motion $y(t)$ of the base from its static equilibrium position is shown in Fig. 2. The corresponding time displacement of the mass m from its static equilibrium position is denoted $x(t)$. The ability of the mass-spring damper system to damp mechanical vibration is characterized by the displacement transmissibility T_d [1, 8, 9], which is given by the equation (1):

$$T_d = \frac{X}{Y} = \sqrt{\frac{k^2 + (c \cdot \omega)^2}{\left(k - m \cdot \omega^2\right)^2 + (c \cdot \omega)^2}} = \sqrt{\frac{1 + (2\zeta \cdot r)^2}{\left(1 - r^2\right)^2 + (2\zeta \cdot r)^2}} \; [-], \tag{1}$$

Where:

X…Displacement amplitude of the response x [m],

Y…Displacement amplitude of the base [m],

k…Spring stiffness [N·m^{-1}],

c…Viscous damping coefficient [N·s·m^{-1}],

ω…Frequency of oscillation [rad·s^{-1}],

m…Mass [kg],

ζ…Damping ratio [–],

r…Frequency ratio [–], which is defined by the equation (2):

$$r = \frac{\omega}{\omega_n} = \omega \cdot \sqrt{\frac{m}{k}} = 2\pi \cdot f \cdot \sqrt{\frac{m}{k}} \; [-], \tag{2}$$

Where:

ω_n…Natural frequency [rad·s^{-1}],

f…Frequency [Hz].

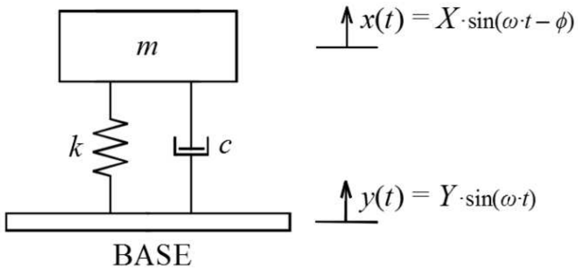

Fig. 2 *Schematic diagram of a spring-mass damper system subjected to base motion*

In the case of the harmonically excited vibration, it is also possible to express the displacement transmissibility as follows (see Fig. 1):

$$T_d = \frac{v_O}{v_I} = \frac{a_O}{a_I} \; [-], \tag{3}$$

Where:

v_O…Velocity amplitude on the output side of a harmonically excited system [m·s^{-1}],

v_I… Velocity amplitude on the input side of a harmonically excited system [m·s^{-1}],

a_O…Acceleration amplitude on the output side of a harmonically excited system [m·s^{-2}],

a_I… Acceleration amplitude on the input side of a harmonically excited system [m·s^{-2}].

Based on the displacement transmissibility value, there are three different types of mechanical vibration:

- damped vibration ($T_d < 1$),
- undamped vibration ($T_d = 1$),
- resonance vibration ($T_d > 1$).

Under the condition $dT_d/d\zeta = 0$ in the Eq. (1), it is possible to determine the frequency ratio r_0 at which the displacement transmissibility has its maximum:

$$r_0 = \frac{\sqrt{\sqrt{1 + 8\zeta^2} - 1}}{2\zeta} \; [-]. \tag{4}$$

It is evident from the equation (4) that the frequency ratio r_0 generally decreases with the increasing damping ratio ζ. The phase angle ϕ (see Fig. 2) of the response $x(t)$ with regard to the base motion $y(t)$ under the harmonic base excitation [1] is expressed by the equation (5):

$$\phi = \tan^{-1}\left[\frac{m \cdot c \cdot \omega^3}{k \cdot \left(k - m \cdot \omega^2\right) + (c \cdot \omega)^2}\right] = \tan^{-1}\left[\frac{2\zeta \cdot r^3}{1 + \left(4\zeta^2 - 1\right) \cdot r^2}\right] [rad] \tag{5}$$

2 Experimental

2.1 Measurement methodology of displacement transmissibility

Experimental measurements of frequency dependencies of the displacement transmissibility were performed by the forced oscillation method at the frequency range of 2-1500 Hz on samples of the ground plane dimensions 60 mm × 60 mm. The measuring equipment (see Fig. 3) consists of BK 3560-B-030 signal pulse multianalyzer, BK 4810 mini-shaker and BK 2706 power amplifier (all from Brüel & Kjær, Denmark).

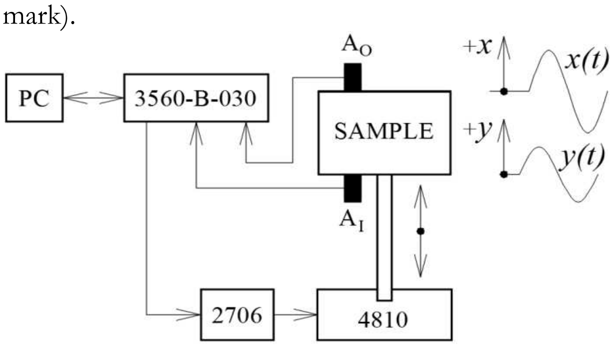

Fig. 3 *Schematic diagram of measuring equipment*

The displacement transmissibility T_d was subsequently determined from the equation (3) by means of the acceleration amplitudes that were experimentally measured by the accelerometers A_I and A_O (BK 4393). Each measurement was repeated 5 times at an ambient temperature of 23 °C.

2.2 Investigated multilayer structures

As mentioned above, the vibration damping properties were investigated for multilayer structures which consisted of different material layers. The specification of basic materials, which were used in tested multilayer structures, is shown in Tab. 1.

Tab. 1 *Specification of basic materials*

Material designation	Material characterization	Density ρ [kg·m^{-3}]
CON	Concrete	2205
CORK	Cork	204
PURL	Low-density open-cell polyurethane foam	150
PURM	Middle-density open-cell polyurethane foam	400
PURH	High-density open-cell polyurethane foam	510
PVC	Polyvinyl chloride	228
RUB	Rubber	1039
STE	Steel	7850
RRUB	Recycled granulated rubber	675
RPUR	Recycled polyurethane foam	84
RTEX	Recycled textile material	77

3 Measured results and discussion

This chapter deals with different factors that have influence on the displacement transmissibility T_d of investigated multilayer structures.

3.1 Displacement transmissibility of basic materials

The frequency dependencies of the displacement transmissibility of certain basic materials, which are applied in multilayer composite structures, are shown in Fig. 4. The inset legends indicate the sample type (see Tab. 1) and its thickness (behind the sample type) in millimetres. It is visible that the rigid materials (see Fig. 4a) embody generally lower vibration damping properties (i.e. higher values of the displacement transmissibility) compared to the soft materials (see Fig. 4b). Therefore rigid materials are not suitable in order to damp mechanical vibration. In many cases (e.g. for mounting a machine to a base), it is desirable to ensure not only the vibration elimination of a system, but also its rigidity. For this reason it is necessary to design a suitable multilayer composite structure.

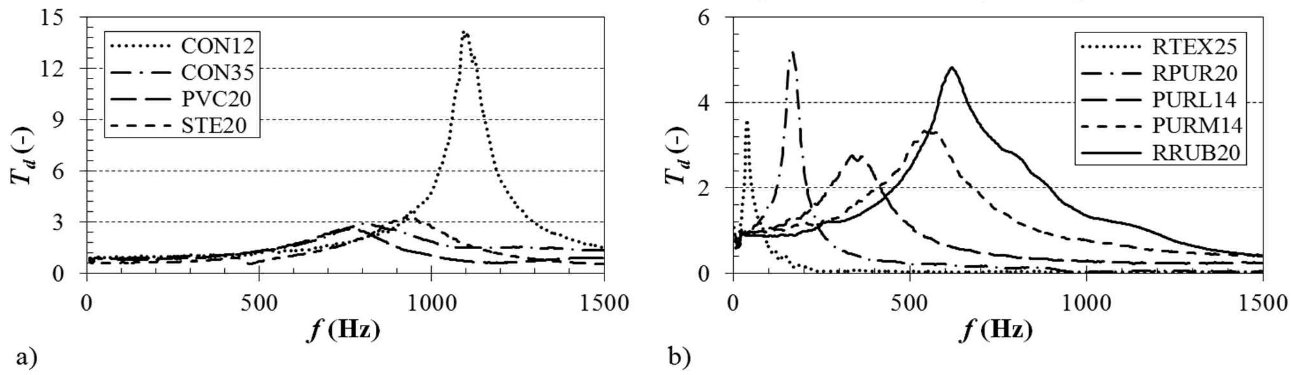

Fig. 4 *Frequency dependencies of the displacement transmissibility for rigid (a) and soft (b) materials*

3.2 Displacement transmissibility of multilayer structures

In this chapter, the designation principle of the investigated multilayer structures is performed as follows: The first material type (incl. its thickness in mm) in the name of a given multilayer composite structure is connected to the input side I (see Fig. 1), i.e. to the BK 4810 mini-shaker (see Fig. 3). The last material (incl. its thickness in mm) in the name of a given multilayer structure is on the output side O (see Fig. 1) of a harmonically excited system. As indicated in Fig. 1, the mechanical vibration is gradually propagated from the input side I through other materials (from the left in the name of a given multilayer structure) to the output side O.

The frequency dependencies of the displacement transmissibility T_d of single- and two-layer structures are compared in Fig. 5. Vibration damping properties are characterized by the first resonance frequency f_{R1} at which the first maximum displacement transmissibility value T_{dmax} is reached ($f_{R1} \approx T_{dmax}$). The first resonance frequency peak position of the tested two-layer composite structures is shifted to lower excitation frequencies compared to the single-layer basic materials. For this reason the presented two-layer composite structures have a better ability to damp mechanical vibration.

It is visible (see Fig. 5a) that the material thickness has a positive influence on vibration damping. It is caused by a higher inertial friction of the thicker recycled polyurethane foam (t = 30 mm) during propagation of mechanical vibrations through the material structure under dynamic loading compared to the recycled polyurethane foam of the thickness t = 10 mm. This phenomenon is observed mainly at lower excitation frequencies (i.e. f < 500 Hz).

It is also evident that the inertial mass, in this case the concrete (see Fig. 5a) and the steel (see Fig. 5b), has a positive effect on reduction of mechanical vibrations. In general, a higher inertial mass leads to a decrease of the natural frequency ω_n [10, 11] that is proportional to the square root of the ratio of the material stiffness k to the mass m as indicated in equation (2). This fact is reflected in a decrease of the above-mentioned frequency ratio r_0 and the first resonance frequency f_{R1}.

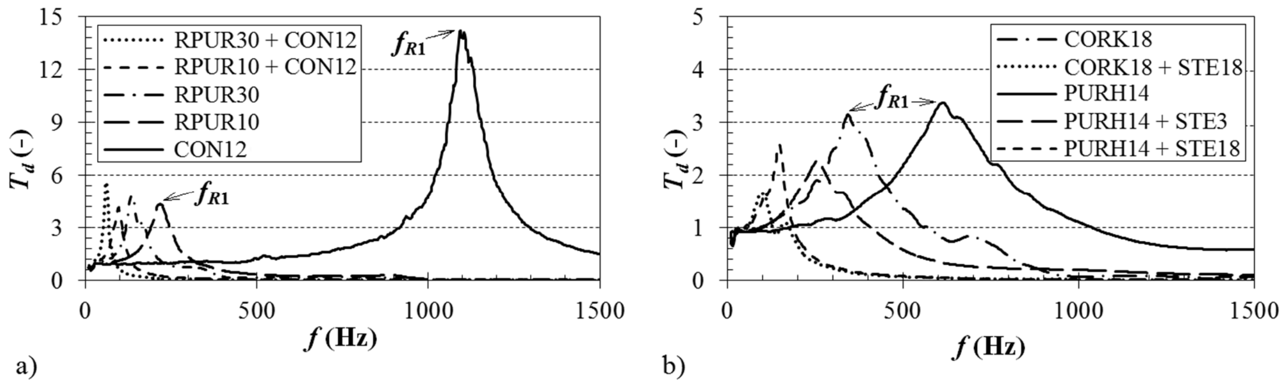

Fig. 5 *Frequency dependencies of the displacement transmissibility of single- and two-layer structures*

Fig. 6 shows the vibration damping ability of different types of middle material layers that were placed between two certain rigid material layers in the investigated three-layer structures. It is evident that vibration damping properties of the tested three-layer structures are generally increasing with decreasing the density of the middle-layer material. Therefore the application of a low-density middle-layer material leads to a decrease in total stiffness of the three-layer structures and also to a shift of the first resonance frequency peak position to lower excitation frequencies. This is particularly evident in the application of recycled low-density materials such as the recycled polyurethane foams and textiles. The application of recycled waste materials can contribute to environmental protection in different areas. On the contrary, the application of tougher materials (e.g. the high-density polyurethane foam in Fig. 6a) in the three-layer structures leads to a lower ability to reduce mechanical vibrations. It is again evident (see Fig. 6b) that the thickness of the applied middle-layer material (i.e. the recycled polyurethane foam) has a positive influence on vibration damping of the tested three-layer structure.

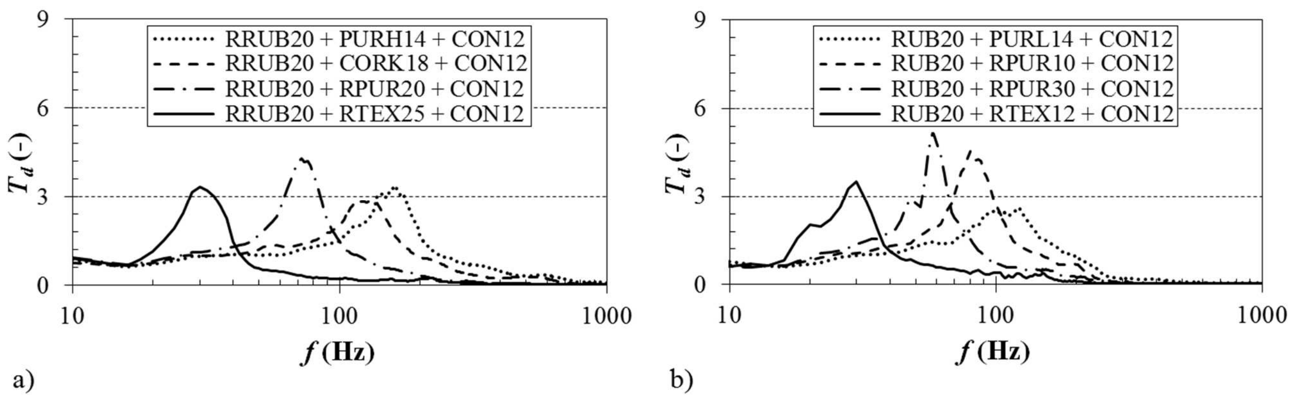

Fig. 6 *Frequency dependencies of the displacement transmissibility of three-layer structures*

The frequency dependencies of the displacement transmissibility T_d of selected four-layer structures are shown in Fig. 7. It is visible that better vibration damping properties are generally obtained by applying softer (i.e. the recycled polyurethane and textile materials in Fig. 7a) and thicker (i.e. the recycled polyurethane foam in Fig. 7b) materials in the tested four-layer structures.

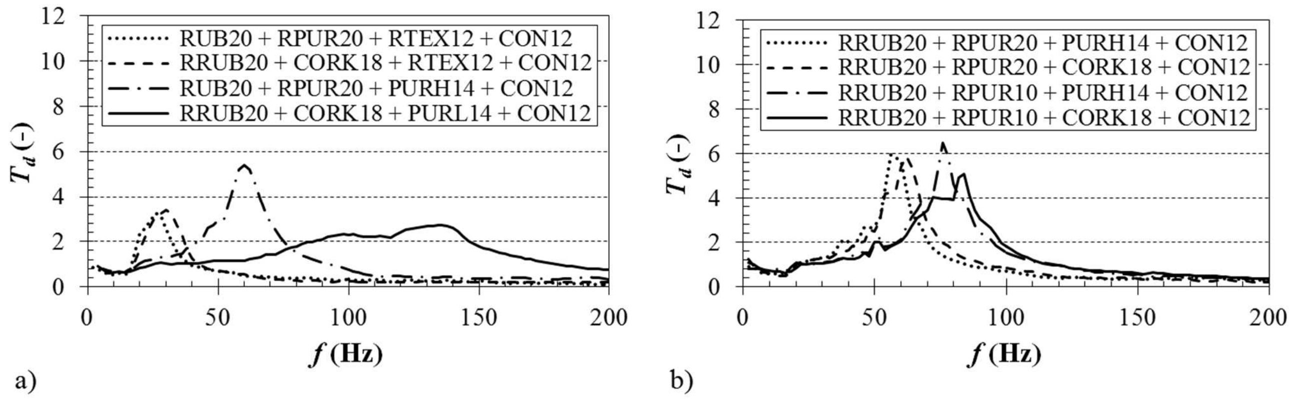

Fig. 7 *Frequency dependencies of the displacement transmissibility of four-layer structures*

3.3 Effect of number of material layers

The effect of number of material layers in multilayer composites structures on the displacement transmissibility is demonstrated in Fig. 8. It is evident that the increasing number of the applied material layers (see Figs. 8a and 8b) in the multilayer composites structures has a positive influence on vibration damping in these cases. The increasing number of the material layers results in a stiffness decrease of the multilayer structures and subsequently leads to a shift of the first resonance frequency peak position to lower excitation frequencies.

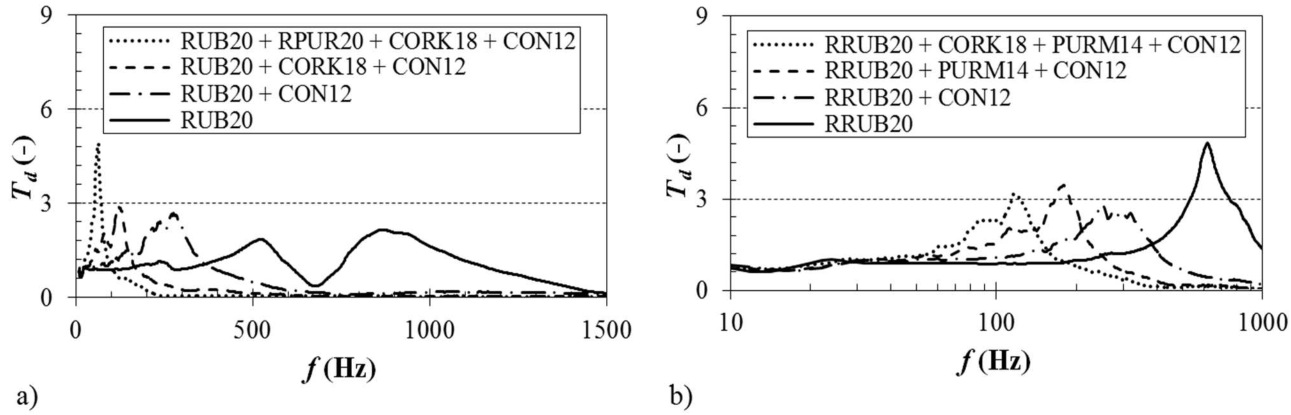

Fig. 8 *Effect of number of material layers on the displacement transmissibility*

3.4 Effect of excitation frequency

The excitation frequency f of mechanical vibration belongs to significant parameters that have an influence on the displacement transmissibility of the investigated multilayer composite structures. Therefore the applicable frequency has an important influence on vibration damping. It can be concluded (see Fig. 4 ÷ Fig. 8) that the resonance vibration (for $T_d > 1$) was generally observed at low excitation frequencies. In the case of some basic rigid materials (e.g. the concrete), the resonance vibration was observed practically in the whole measuring frequency range. Contrariwise, the investigated soft materials (e.g. the recycled polyurethane and textile materials) are characterized by the resonance vibration at significantly lower excitation frequencies (see Fig. 4b). It was also found that the resonance vibration can be eliminated by suitable multilayer composite structures (see Fig. 5 ÷ Fig. 8) compared to the basic single-layer materials.

4 Conclusion

The purpose of this paper was to investigate different factors that have influence on vibration damping properties of multilayer composite structures. It was found in this study that the ability to damp mechanical vibration of the tested multilayer structures is significantly affected by composition of the structures, number of material layers, excitation frequency, material thickness, density and inertial mass. It can be concluded that vibration damping properties of the investigated multilayer composite structures are generally increasing with increasing the excitation frequency, the material thickness and the inertial mass and with decreasing the material density. These factors are reflected in a shift of the first resonance frequency peak position to lower excitation frequencies. In addition, when applying suitable materials, a more efficient vibration isolation system can be also achieved with a higher number of layers in the investigated multilayer structures. Moreover, it is possible to apply suitable recycled materials (e.g. soft polyurethane foams and textile materials) in order to damp mechanical vibration, which can contribute to environmental protection.

Acknowledgements

This work was supported by the European Regional Development Fund in the Research Centre of Advanced Mechatronic Systems project, project number CZ.02.1.01/0.0/0.0/16_019/0000867 within the Operational Programme Research, Development and Education.

References

[1] RAO, S. (2004). Mechanical Vibrations. p. 1078. Pearson Education Inc., New Jersey. USA.

[2] KLIMENDA, F., SOUKUP, J., STERBA, J. (2019). Noise and Vibration Analysis of Conveyor Belt. In: *Manufacturing Technology*. Vol. 19, No. 4, pp. 604–608. Elsevier. Netherlands.

[3] HANZL, P., ZETKOVA, I. (2019). Benefits of a New Approach to Designing Milling Cutter Using Metal Additive Manufacturing. In: *Manufacturing Technology*. Vol. 19, No. 3, pp. 385–390. Elsevier. Netherlands.

[4] SEGLA, S., KAMPO, J. (2018). The Role of Modelling of Road Unevennesses in Vehicle Dynamics. In: *Manufacturing Technology*. Vol. 18, No. 1, pp. 124–129. Elsevier. Netherlands.

[5] DIZO, J., BLATNICKY, M. (2019). Evaluation of Vibrational Properties of a Three-wheeled Vehicle in Terms of Comfort. In: *Manufacturing Technology*. Vol. 19, No. 2, pp. 197–203. Elsevier. Netherlands.

[6] KORMANIKOVA, E., KOTRASOVA, K. (2017). Buckling and free vibration analysis of sandwich plate. In: *International Journal of Mechanics*. Vol. 11, pp. 299–304. Nord Atlantic University Union. Canada.

[7] BSW (2019). Vibration Insulation. [online]. Available from: https://www.berleburger.com/en/regufoam-vibration-isolation/

[8] CARRELLA, A., BRENNAN, M. J., WATERS, T.P., LOPES, V. Jr. (2012). Force and displacement transmissibility of a nonlinear isolator with high-static-low-dynamic stiffness. In: *International Journal of Mechanical Sciences*. Vol. 55, No. 1, pp. 22–29. Elsevier. England.

[9] LATIF, N. A., RUS, A. Z. M. (2013). Vibration Transmissibility Study of High Density Solid Waste Biopolymer Foam. In: Proceedings of the 2nd International Conference on Mechanical Engineering Research, ICMER 2013, pp. 1 – 9. Kuantan, Malaysia.

[10] BELHAQ, M., BICHRI, A., HOGAPIAN, J.D., MAHFOUD, J. (2011). Effect of electromagnetic actuations on the dynamics of a harmonically excited cantilever beam. In.: *International Journal of Non-Linear Mechanics*. Vol. 46, No. 6, pp. 828–833. Elsevier. England.

[11] DU TOIT, N.E., WARDLE, B.L., KIM, S.G. (2005). Design considerations for MEMS-scale piezoelectric mechanical vibration energy harvesters. In.: *Integrated Ferroelectrics*. Vol. 71, pp. 121–160. TAYLOR & FRANCIS LTD. England.