Mechanical modelling and application of vibroacoustic isolators in railway tracks

Artur Zbiciak[1], *Cezary* Kraśkiewicz [1,*], *Wojciech* Oleksiewicz[1], *Monika* Płudowska-Zagrajek[1] and *Cezary* Lipko[2]

[1]Warsaw University of Technology, Faculty of Civil Engineering, Institute of Roads and Bridges, 16 Armii Ludowej Ave., 00-637 Warsaw, Poland
[2]TINES Capital Group SA, TINES Railway, 23 Ciepłownicza St., 31-574 Cracow, Poland

Abstract. The paper presents systematization and description of vibroacoustic isolators used in railway tracks (due to track structure type), with special attention paid to resilient mats. As in the second part of the paper the state-space mechanical model of a system with Under-Ballast Mat is formulated. Also some numerical problems arising from the mass matrix singularity are discussed. The poles of the system were calculated by using Matlab. Moreover, the influence of various parameters on the system's insertion loss and its transmissibility was visualized in figures.

1 Introduction

Rail track structure consists of many components, with different functions and material characteristics. As rail traffic generates dynamic effects, such as vibrations and structure-borne noise, during rail vehicle movement, it is appropriate to use additional elements to reduce those negative effects called vibroacoustic isolators. Due to track structure type (ballasted or ballastless track system), elements location in the track structure or material characteristics, we can identify different types of isolators (i.e. rail pads, under sleeper pads, under-ballast mats).

2 Application areas of vibroacoustic isolators in railway tracks

Classifying kinds of rail track superstructures it is appropriate to assume as the main criterion occurrence in a given structure of the ballast, as the main superstructure layer and its function of shaping the geometry of the track. If this feature is fulfilled, track structure is qualified as ballasted track system (Fig. 1). Otherwise, it is qualified as ballastless track system (Fig. 2). It is important for ballast layer to meet both of the above conditions, because ballast layer can occur in ballastless track system - in that case it has different functions i.e.: drainage function, thermal insulation function or noise reduction function.

* Corresponding author: c.kraskiewicz@il.pw.edu.pl

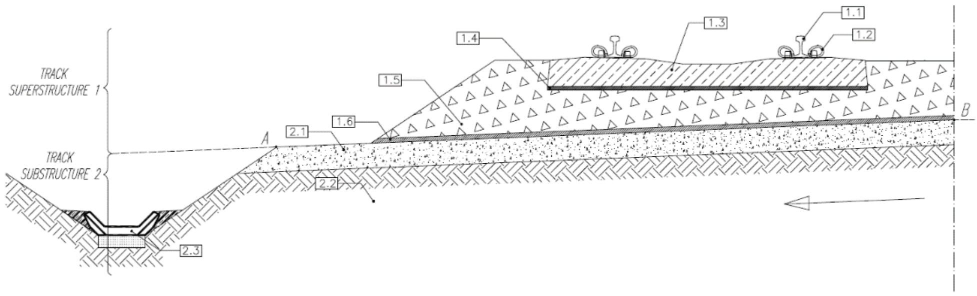

Fig. 1. Components of ballasted track system. Key: Track superstructure (1): 1.1 – Vignole rail profile, 1.2 – rail fastening system (type SB or W14), 1.3 – rail sleeper, 1.4 – (option) under sleeper pad, 1.5 - ballast, 1.6 – (option) under-ballast mat. Track substructure (2): 2.1 – blanket layer, 2.2 – subgrade, 2.3 – surface drainage. Note: In case of use of under-ballast mat (UBM), no under sleeper pads (USP) is applied.

The following figure (Fig. 2) shows a use of a ballastless track system in a tunnel on example of the Warsaw Metro Line II.

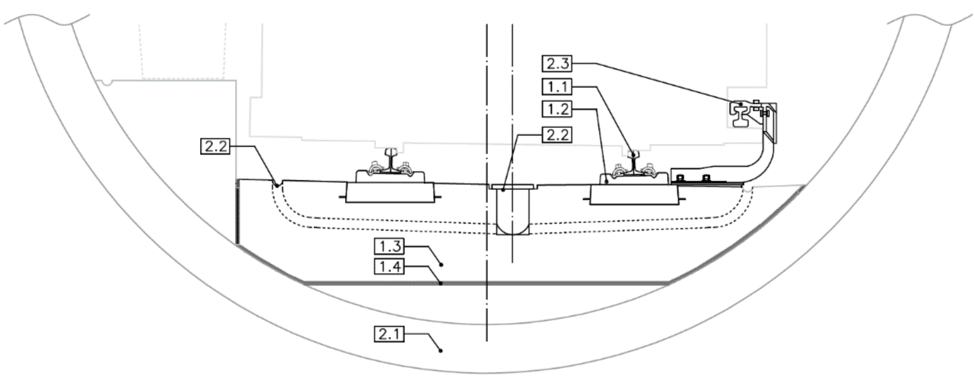

Fig. 2. Components of ballastless track system in a tunnel on example of the Warsaw Metro line II. Key: 1.1 – Vignole rail profile type 49E1; 1.2 – rail fastening system – Embedded Blocks System (EBS); 1.3 – concrete track slab; 1.4 – slab-track mat; 2.1 – concrete inverted slab; 2.2 – surface drainage; 2.3 – conductor-rail.

Application areas of vibroacoustic isolators depend on type of rail track structure, rail fastening system or theirs functions. At Fig. 3 are shown components of vibroacoustic isolators system and their main application areas:

1. resilient fastening (fixing) of rail foot: resilient clamps or grouting;
2. resilient fastening (supporting) of rail foot: rail pad, continuous rail pad, discrete or continuous grouting;
3. resilient fixing in indirect rail fastening system: tension clamps, anchor bolts with compression spring;
4. resilient support of rail supporting structure in indirect rail fastening system: base plate pad, discrete grouting;
5. resilient support or fixing of rail supporting structure (sleeper or block): under sleeper pad, under block pad, grouting;
6. resilient fixing and support of railway track: under-ballast mats or slab-track mats – which with rail track structure compose floating slab track system;

7. filling gaps between fishplate and rail web (noise reduction): grouting or rail web filler block.

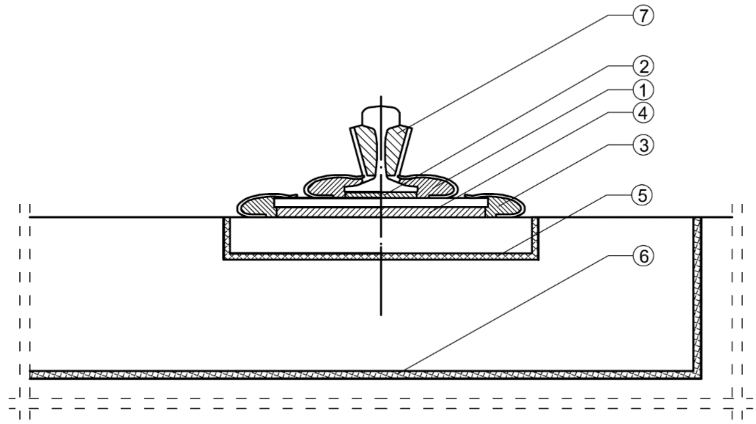

Fig. 3. Application areas of vibroacoustic isolators in railway tracks.

Resilient mats can be grouped into two categories, because of the range of applications in various types of railways track construction:
a) Under-Ballast Mats (UBM)/ Sub-Ballast Mats (SBM) or ger. Unterschot-termatten (USM) – used in ballast railway track (Fig. 1); among which two varieties can be classified due to the main purpose of their use:
- used primarily for isolation from vibration,
- used primarily for stress reduction in ballast.
b) Slab-Track Mats (STM) – used in ballastless railway track (applied under or from the sides of concrete track base plate – Fig. 2), with three varieties of slab support system according to standard [4]:
- discrete support (steel springs or elastomeric pads),
- linear support (strip mats),
- continuous support (elastomeric mats) – Fig. 2.

Characteristics of resilient mats important for vibration isolation and their test methods were given in standard [3, 4] and [8, 9, 10].

The further part of the paper will address mechanical model of ballasted track system with Under-Ballast Mat.

3 Mechanical model

Mechanical model of the system with Under-Ballast Mat (UBM) is shown in Fig. 4. Such a model was partially analysed in [15] and is also presented in Standard [2] for calculation of insertion loss. The objective of this analysis is to present the state-space formulation of differential equations representing this model.

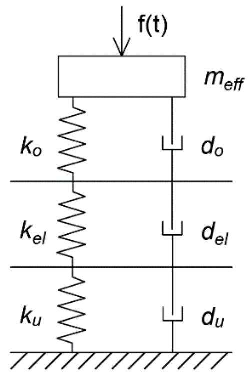

Fig. 4. Mechanical model of a system with Under-Ballast Mat.

Material parameters of mechanical model shown in Fig. 4 are presented in Table 1. Let us note that instead of damping coefficients d_{o} and d_{el}, Table 1 gives loss factors η_{o} and η_{el}. Further analysis will be conducted assuming $\eta = 2\xi$ (see Standards [1-4]), where $\xi = \frac{d}{2\sqrt{km}}$ denotes damping ratio [5]. Thus, damping coefficients can be calculated as follows $d = \eta\sqrt{km}$.

Table 1. Parameters of mechanical model shown in Fig. 4.

m_{eff} [kg]	k_{o} [MN/m]	k_{el} [MN/m]	k_u [MN/m]	η_{o} [−]	η_{el} [−]	d_u [MNs/m]
2600	220	140	1500	0,35	0,05	1,2

Mechanical model representing a rail track system with vibroacoustic elements can be described by inertia $\mathbf{M}$, damping $\mathbf{C}$ and stiffness $\mathbf{K}$ matrices, as follows

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{f} \tag{1}$$

where $\mathbf{f}$ represents loading vector. In case of analysed system, we obtain

$$\mathbf{M} = \begin{bmatrix} m_{eff} & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}, \quad \mathbf{K} = \begin{bmatrix} k_{\mathrm{o}} & -k_{\mathrm{o}} & 0 \\ -k_{\mathrm{o}} & k_{\mathrm{o}} + k_{el} & -k_{el} \\ 0 & -k_{el} & k_{el} + k_u \end{bmatrix}, \quad \mathbf{C} = \begin{bmatrix} d_{\mathrm{o}} & -d_{\mathrm{o}} & 0 \\ -d_{\mathrm{o}} & d_{\mathrm{o}} + d_{el} & -d_{el} \\ 0 & -d_{el} & d_{el} + d_u \end{bmatrix} \tag{2}$$

In many cases the system of 2nd order differential equations (1) should be represented in the following state-space form

$$\begin{aligned} \dot{\mathbf{y}} &= \mathbf{A}^s\mathbf{y} + \mathbf{B}^s\mathbf{u} \\ \mathbf{z} &= \mathbf{C}^s\mathbf{y} + \mathbf{D}^s\mathbf{u} \end{aligned} \tag{3}$$

where $\mathbf{A}^s$ denotes the system matrix, $\mathbf{B}^s$ is the input matrix, $\mathbf{C}^s$ is the output matrix and $\mathbf{D}^s$ is the feed through matrix. Comparing Eq. (1) and Eq. (3) gives

$$\mathbf{y}=\begin{bmatrix}\mathbf{x}\\ \dot{\mathbf{x}}\end{bmatrix},\quad \mathbf{A}^s=\begin{bmatrix}\mathbf{0} & \mathbf{I}\\ -\mathbf{M}^{-1}\mathbf{K} & -\mathbf{M}^{-1}\mathbf{C}\end{bmatrix},\quad \mathbf{B}^s=\begin{bmatrix}\mathbf{0}\\ \mathbf{M}^{-1}\end{bmatrix},\quad \mathbf{C}^s=\begin{bmatrix}\mathbf{C}^{ox} & \mathbf{C}^{ov}\end{bmatrix},\quad \mathbf{D}^s=\mathbf{0} \tag{4}$$

The inertia matrix shown in Eq. (2) is singular. On the other hand, in order to obtain the system matrix $\mathbf{A}^s$ applying Eq. (4), needs the mass matrix to be inverted. The reason of such a problem is the presence of degrees of freedom without inertia, so called half degrees of freedom. This problem can be solved by using the method presented in [11], where a very useful and easy to be implemented in a computer routine algorithm was invented. This algorithm was programmed in Matlab, giving the following results for the system matrix and the input matrix

$$\mathbf{A}^s=\begin{bmatrix}0 & 0 & 0 & 1\\ 747.96 & -1212.26 & 339.25 & 0.90\\ 18.34 & 84.08 & -1324.83 & 0.02\\ -8465.30 & -38805.52 & 34538.85 & -10.19\end{bmatrix},\quad \mathbf{B}^s=\begin{bmatrix}0 & 0 & 0\\ 0 & 3.40 & 0.08\\ 0 & 0.08 & 0.82\\ 384.62 & 346.14 & 8.49\end{bmatrix} \tag{5}$$

Using the functions implemented in Control System Toolbox of Matlab software, the poles of analysed system were calculated (see Fig. 5). Poles show the frequencies where the system will amplify inputs. The poles depend only on the distribution of mass, stiffness and damping throughout the system. They do not depend on where the forces are applied or where displacements are measured.

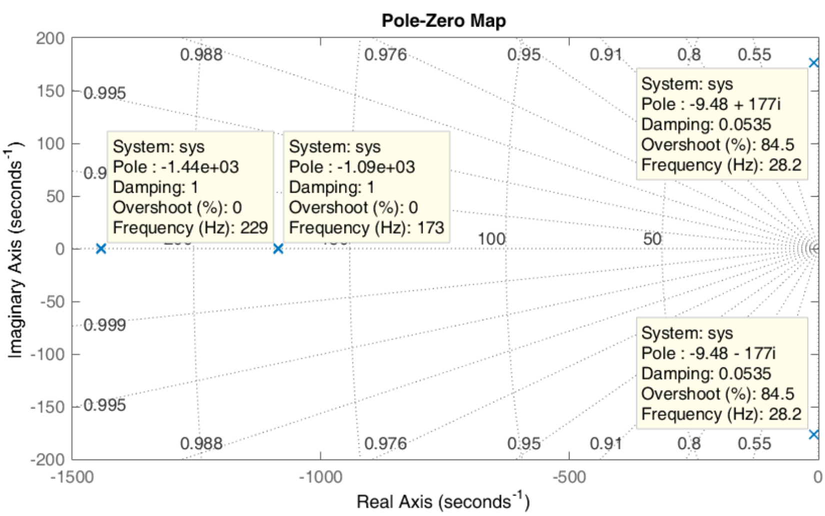

Fig. 5. Pole-Zero Map of analysed dynamic system.

Let us analyse additional properties of mechanical system shown in Fig. 4. Paper [15] presents formulas to calculate transmissibility and insertion loss of such system. Using these formulas we will show the influence of certain parameters on its dynamic properties.

Fig. 6 visualizes insertion loss diagrams for various mass parameters m_{eff}. Horizontal axis used in Fig. 6 (also in Fig. 7) represents centre frequencies of 1/3 octave bands. The plots depicted in Fig. 6 indicate regions of vibration effectiveness where $IL > 0$ dB. Increasing the effective mass results in increase of vibration effectiveness region.

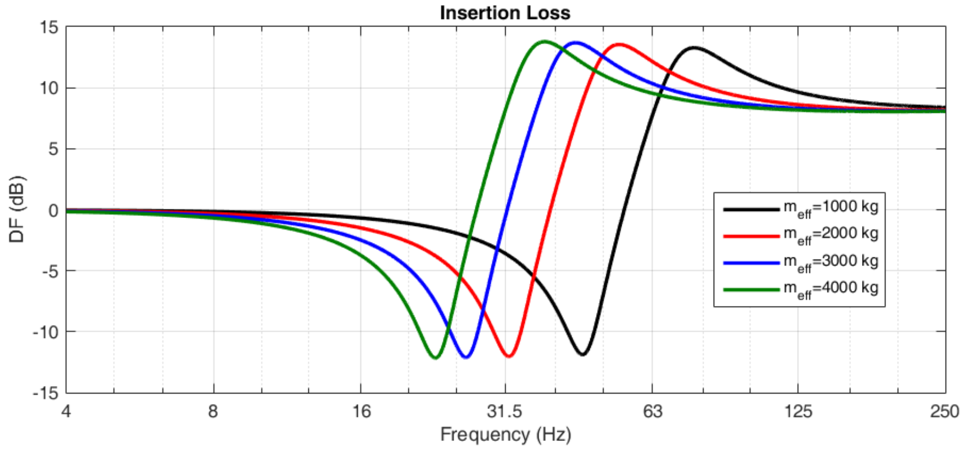

Fig. 6. Insertion loss diagrams for various effective mass parameters.

Insertion loss diagrams for various mat stiffness parameters k_{el} are given in Fig. 7. Increase of mat's stiffness makes the effectiveness region smaller and decreases the amplitudes of insertion loss.

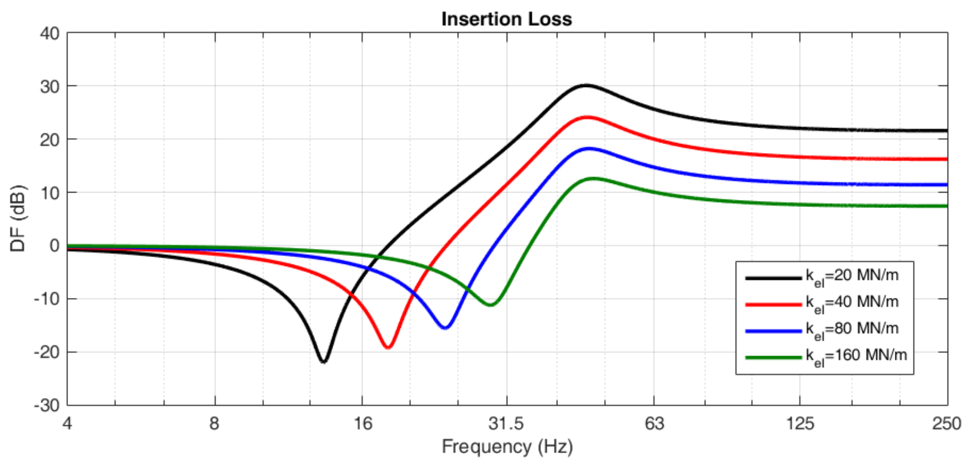

Fig. 7. Insertion loss diagrams for various mat stiffness parameters.

Finally, Fig. 8 visualizes transmissibility of the system for various mat loss factors η_{el} . Such diagrams are presented in many papers [12-14] for various mechanical models representing railway track systems with vibroacoustic isolators.

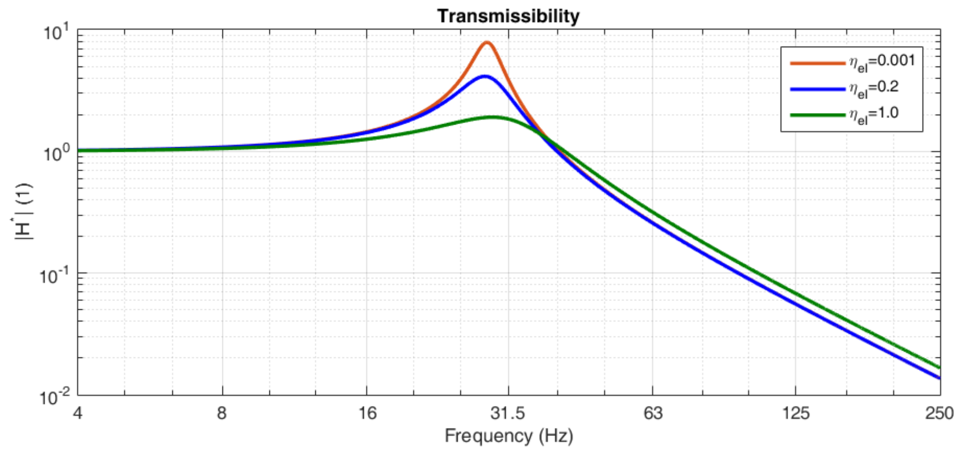

Fig. 8. Transmissibility diagram for various loss factors.

4 Conclusions

Vibroacoustic isolators are components used to limit the negative effects (vibration and noise) generated by rail traffic on the track, and vary depending on material characteristics, application area or due to track structure type (ballasted or ballastless track system). One of goals of this paper was to systematize vibroacoustic isolators used as close as possible to the place of vibration generation during the movement of rail vehicles. As there are reasonable grounds to conclude that those elements will find increasingly widespread use in rail track structure along with further advances in engineering and material engineering, and taking into consideration increased requirements for protecting surroundings of transport routes from negative transport effects.

The results of calculations presented in this paper were obtained by applying a mechanical system containing half degrees of freedom. It results in singularity of inertia matrix. The problem was solved using a method proposed in literature [11]. It would be possible to use the same methodology in case of multi degree of freedom linear models representing rail track system with resilient elements. In such systems, half degrees of freedom can be associated with more general models of vibroacoustic elements, described by fractional-order derivatives [6], as it was suggested in [15]. Such an approach would make possible to fit better experimental results. More advanced models of railway tracks systems with vibroacoustic isolators can be formulated by using Finite Element Method [7].

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