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Measuring vibration damping level on conventional rail track structures

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Subject review

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Measuring vibration damping level on conventional rail track structures

The rail-track vibration damping level, as one of key properties for determining proportion of rail track influence in the total rail traffic noise and vibration levels, is estimated in the paper. A detailed overview of research conducted so far on dynamic properties of rail tracks is given, and the methods presented are compared on two test sections. The data processing method, various accelerometer positons, and various types of modal hammers as excitation sources, are also considered in the analysis of measurement results. The results obtained by various analyses are compared to each other so as to increase accuracy in the determination of the vibration damping level.

Key words:
vibrations, vibration damping level, dynamic properties of a rail track structure, typical noise measurements

Pregledni rad

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Mjerenje stupnja prigušenja vibracija na klasičnim kolosiječnim konstrukcijama

U radu se ocjenjuje stupanj prigušenja vibracija na kolosijecima, jednog od ključnih svojstava pri određivanju utjecaja kolosijeka u ukupnim razinama buke i vibracija od prometovanja tračničkih vozila. Prikazan je i detaljan pregled znanstvenih istraživanja na temu dinamičkih svojstava kolosijeka, a navedene metode uspoređene su na dvije ispitne lokacije. Nadalje, u analizi rezultata mjerenja razmatra se metoda obrade podataka, različite pozicije akcelerometara i različiti tipovi modalnih čekića kao izvora pobude. Rezultati dobiveni različitim analizama međusobno su uspoređeni u svrhu što preciznijeg određivanja stupnja prigušenja vibracija.

Ključne riječi:
vibracije, stupanj prigušenja vibracija, dinamička svojstva kolosiječne konstrukcije, tipska mjerenja buke

Übersichtsarbeit

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Messungen des Dämpfungsgrades bei Vibrationen klassischer Gleiskonstruktionen

In dieser Arbeit wird der Vibrationsdämpfungsgrad bei Gleiskonstruktionen beurteilt. Dabei handelt es sich um einen der entscheidenden Parameter zur Ermittlung des Einflusses der Gleise auf den gesamten durch Eisenbahnverkehr entstehenden Lärm- und Vibrationspegel. Eine detailierte Übersicht wissenschaftlicher Untersuchungen bezüglich dynamischer Gleiseigenschaften wird dargestellt. Die angegebenen Methoden werden für zwei Testorte gegenübergestellt. Des Weiteren werden bei der Analyse der Resultate verschiedene Bearbeitungsmethoden, Positionen der Beschleunigungsmesser und Modalhammertypen als Anregungsquelle betrachtet. Die Resultate verschiedener Analysen werden gegenübergestellt, um eine präzise Ermittlung des Vibrationsdämpfungsgrades zu ermöglichen.

Schlüsselwörter:
Vibrationen, Vibrationsdämpfungsgrad, dynamische Eigenschaften von Gleiskonstruktionen, typische Lärmmessungen

1. Introduction

The interaction between rail vehicle wheels and underlying rails results in the noise and vibrations that exert a negative influence on people and structures situated in the immediate vicinity of rail tracks. This problem is most acute in urban communities, and the systematic resolution of the noise and vibration issue starts with the detection of source, and with the determination of noise and vibration propagation intensity and related mechanisms.

1.1. Noise and vibrations generated by rail traffic

Since the very beginnings of rail transport, the use of rail vehicles is accompanied by inevitable generation of noise and vibrations. The issue of noise and vibrations started to be addressed in a systematic manner in the 1960s when the noise was characterised as a source of uneasiness in residential districts [1]. Thus the growing resistance to introduction of rail lines is explained by the noise generated during operation of rail traffic. The Regulation (EC) 1304/2014 of the European Commission and of the Council [2], which is currently in force in the European Union, specifies maximum noise levels that rail vehicles are allowed to emit when out of operation, when starting to operate, when operating at constant speed, as well as noise levels in the driver's cabin. Consequently, all new and renovated railway vehicles intended to operate along the EU railway system must meet the criteria specified in the Regulation, Table 1.

Table 1. Limit noise level for vehicles operating at constant speed [2]

Category of the rolling stock subsystem	$L_{pAeq, Tp (80 km/h)}$ [dB]	$L_{pAeq, Tp (250 km/h)}$ [dB]
Electric locomotives and OTMs with electric traction	84	99
Diesel locomotives and OTMs with diesel traction	85	N.a.
EMUs	80	95
DMUs	81	96
Cocaches	79	N.a.
Wagons (normalised to APL = 0,225)	83	N.a.

The regulation also calls for application of the standard HRN EN ISO 3095:2013 [3] according to which the A-weighted equivalent continuous sound pressure level must be evaluated at the speeds of 80 km/h and 250 km/h. The Regulation [2] specifies requirements for noise levels of railway vehicles, and the testing is conducted on a reference track defined in section 6.2 of EN ISO 3095:2013 [3]. However, it is also allowed to conduct the test on the track that does not comply with reference track conditions in terms of acoustic rail roughness level and track decay rate as long as the noise levels measured according to Section 6.2.2.3.2 do not exceed the limit values set in Section 4.2.3. The **acoustic rail roughness level** (standard HRN EN 15610:2009 [4]) and the **track decay rates** (standard HRN EN 15461:2011 [5]) shall be determined in any case. If the track on which the tests are performed does meet the reference track conditions, the measured noise levels shall be marked "comparable", otherwise they shall be marked "non-comparable". It shall be recorded in the technical file whether the measured noise levels are "comparable" or "non-comparable".

The measured acoustic rail roughness values of the test track remain valid during a period starting 3 months before and ending 3 months after this measurement, provided that during this period no track maintenance has been performed which influences acoustic roughness of the rail.

The measured track decay rate values of the test track shall remain valid during a period starting one year before and ending one year after this measurement, provided that during this period no track maintenance has been performed which influences the track decay rates.

The determination of these track parameters enables definition of the track share in the total noise levels generated during the passage of trains. Thus, it can be concluded, on sections meeting the mentioned standards, whether the track is sufficiently "quiet" to enable determination of the noise generated by the vehicle itself during typical vehicle noise testing, without the noise caused by an acoustically poor track structure.

The main sources of noise and vibrations are vehicle engines, aerodynamic noise at high speeds of travel, and the contact between the vehicle wheel and the rail. The travelling speed is the main parameter that influences dominant sources of noise, and so the engine noise is dominant at small speeds of up to 20 km/h, the wheel rolling noise is dominant at speeds ranging from 20 km/h to 250 km/h, and the aerodynamic noise becomes dominant as higher speeds. These values are not equal for all vehicles and in all conditions, and so different speed intervals are stated in literature for the wheel rolling noise: 50 – 270 km/h [6, 7], 40 – 250 km/h [8], and 20 – 250 km/h, according to most recent studies [9]. The rolling noise occurs at lower speeds if modern and less noisy engines are used for starting the vehicle, and if the contact surfaces of wheels and rails are rougher.

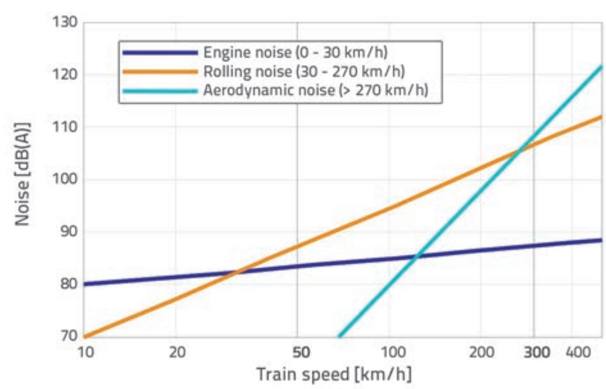

Figure 1. Noise levels for various sources and various speeds of travel [10]

It is precisely this last source that is dominant in urban and suburban traffic, where the speed of travel is not high enough to take into account the aerodynamic component of noise. In urban environments, the noise and vibrations constitute a serious problem for people and structures situated in the vicinity of railway lines. Due to self-weight of vehicles, and dynamic forces resulting from unevenness of the rail and wheel contact zone, the movement of vehicles along rails causes oscillations or vibrations of rail vehicles and track structure. The unevenness of rails with different wavelengths affects the rail infrastructure in a variety of ways. Greater wavelength irregularities cause a significant dynamic stress in rails, deterioration of fastening accessories and ballast, and a premature fatigue of steel at the load-supporting rail surface [11]. A small wavelength corrugation, also known as the acoustic roughness of rails, causes high-frequency vibrations in rails and wheels. At high frequency, the energy of vibrations caused by the wheel and rail contact propagates through the air in form of sound waves (noise), while low frequency vibrations are transferred via rails to the bottom parts of the structure. The frequency of vibrations propagating into the environment due to interaction between rail vehicles and track structures range from 0 to 100 Hz, while airborne sound wave frequencies vary from 30 to 5000 Hz [12].

After it was established that roughness of contact between the wheel and the rail during operation of rail traffic results in oscillations, i.e. in vibrations of wheels and rails, the critical phenomenon has become the propagation of vibrations through the track structure and vehicle components. The oscillation of track and vehicle components due to vibrations caused by the wheel and rail interaction results in emission of sound waves into the air, and these sound waves are perceived by humans as noise. Being the final element with low attenuating properties, the wheel has a clear set of oscillations forms that are highly significant for characterisation of its vibrations. Unlike the wheel, the rail is a non-finite element and as such it allows one or several forms of oscillations and, hence, a good propagation of structural waves. The self-attenuation of vibrations is much greater on the track than on the wheel of the vehicle. Although the track structure induces behaviour that can be described by lower-frequency vibration forms, the rail resonating properties are not similar to the wheel resonating properties. In case of rail, various structural waves can be identified for a single frequency. At lower frequencies, we have vertical and horizontal bending, torsion and longitudinal waves but, at higher frequencies, additional waves occur that also include deformation of the rail cross section [1]. All these rail vibration forms are transferred to the surrounding medium – air – where they create a sound pressure that is perceived by human ear as noise. In addition to being dependent on the speed, rail vibrations generated during operation of traffic also greatly depend on the stiffness of the under rail pads i.e. on the fastening system, and on roughness of the wheel to rail contact zone. The influence of the most significant factors on total noise levels generated by vehicles is shown in Table 2 [13]. The rail temperature, which can differ significantly from air temperature due to generation of heat, can also influence temperature of under rail pads and, hence, their stiffness, which is why rail temperature must be taken into account when estimating stiffness of the under rail pads [14].

The most significant parameters that have to be taken into account to improve repeatability and reproducibility of noise measurements are:

1. **Measured value** in the sense of clear definition of the type of measured noise level, measurement period, and averaging time. A-weighted sound pressure level is appropriate.
2. **Under rail pad behaviour** in the sense of behaviour of materials, load and temperature influence.
3. **Combination of wheels and rails** situated at the section under study and excitation due to roughness that has to be taken into account by measuring A-weighted railhead vibrations. These measurements can easily be repeated by accelerometer, and can also be properly reproduced subject to good determination of rail roughness level.
4. **Effect of sound transfer at measurement section**, especially at the distance of 25 m from the noise source, in combination with the influence of wind and temperature, can reduce repeatability of measurements, and so measurements at 7.5 m from the source are recommended.
5. **Vehicle speed**: greater noise level deviations may occur at greater speeds if speed measurements are not accurate enough [13].

1.2. Vibration and noise measurements

As the dynamic range of spectrum values is very wide, the noise and vibrations are mostly presented on a logarithmic scale expressed in decibels [dB] according to the following expression: 1

$$L\left[dB\right] = 20 log \frac{p_1}{p_2} \tag{1}$$

Table 2. Variation of influence parameters with vehicle noise levels for usual track structures [13]

Parameter	Parameter value for minimum noise level	Parameter value for maximum noise level	Noise level difference for minimum and maximum parameter
Static stiffness of the under rail pad	$5 \cdot 10^9$ [N/m]	$1 \cdot 10^8$ [N/m]	5,9 dB(A)
Pad attenuation	0,5	0,1	2,6 dB(A)
Rail sleeper types	Two-part RC sleepers	Wooden sleepers	3,1 dB(A)
Wheel roughness	Smoothest surface	Roughest surface	8,5 dB(A)
Train speed	80 [km/h]	160 [km/h]	9,4 dB(A)

Power spectrum values are mostly calculated as effective (RMS) values in one-third octave bands. Accelerations are often integrated in the lower frequency range, which is associated with vibrations, so as to obtain vibration velocities [12]. As values are expressed in decibels, they have to be given a reference value. Noise values are expressed in relation to a reference value of $20 \cdot 10^{-6}$ Pa. Human perception of noise is characterised by the A-weighted sound pressure filter. It removes almost all noise frequencies of less than 200 Hz. Noise levels are most often presented with the A filter, and are then marked as dB(A).
The equivalent level of energy $L_{eq,}$ expressed in dB(A), can be calculated by integrating the square sound pressure according to the following formula:

$$L_{eq} = 10log\frac{1}{T}\int_0^T\left[\frac{p_A(t)}{p_0}\right]^2 dt \tag{2}$$

where:
p_0 = $20 \cdot 10^{-6}$ Pa
$p_A(t)$ - A-weighted noise level
T - the integration time [12].

Vibrations are generally defined as an oscillatory motion that can be described by displacement "d" [mm], displacement velocity "v" [mm/s], or acceleration of vibrating body "a" [mm/s²]. Displacement is defined as the distance of the point of interest from the initial position it assumed at the state of no-motion. Velocity is momentary velocity of displacement of the point of interest, and acceleration is the level of change of this velocity. As oscillatory motion is considered in this study, average values of all described parameters are taken to be equal to zero. The relationship between the mentioned values (acceleration, velocity, and displacement) is given in the expression (3), under assumption that vibrations can be expressed as sinusoidal (harmonic) motion of a particle, [15]:

$$a = v(2f\pi) = d(2f\pi)^2 \tag{3}$$

where f is the frequency of vibrations under study. Vibration levels can also be expressed by means of logarithmic scale in decibels [dB], in the way similar to the environmental noise, using expressions (4) and (5), [15]:

$$L_a = 20log\frac{a}{a_0}[dB] \tag{4}$$

$$L_v = 20log\frac{v}{v_0}[dB] \tag{5}$$

where:
L_a - vibration acceleration level in dB
L_v - vibration velocity level in dB
a - vibration acceleration
a_0 - reference acceleration value (10^{-6} according ISO 1683 [16])
v - vibration velocity
v_0 - reference velocity (10^{-9} according ISO 1683 [16]).

2. Overview of current research

The study of noise and vibrations on railways is a very wide and dynamic area of research. The problem of noise and vibrations was first identified very soon after first railway vehicles were put to traffic. The rolling of steel wheels on steel rails is a very strong source of vibrations and, consequently, of noise. The rougher the surface of wheels and rails, the greater the vibrations, and the resulting noise. In order to characterise these phenomena in a systematic manner, and to understand and simulate mechanisms of their occurrence and propagation and, finally, to eliminate, reduce or at least bring down these occurrences to reasonable levels, researchers from all over the world have been conducting extensive research on the issue of noise and vibrations generated by the rolling of rail vehicles.

2.1. Rolling noise

Rolling noise is the most significant source of noise that occurs during operation of railway vehicles. It is generated at the contact between the wheel and the rail, whose interaction causes vibrations. Roughness of wheel and rail running surfaces generates vertical vibrations in wheels and track, depending on their dynamic properties. The mechanism of propagation of vibrations at the wheel to rail contact is shown in Figure 2. Main roughness wavelengths

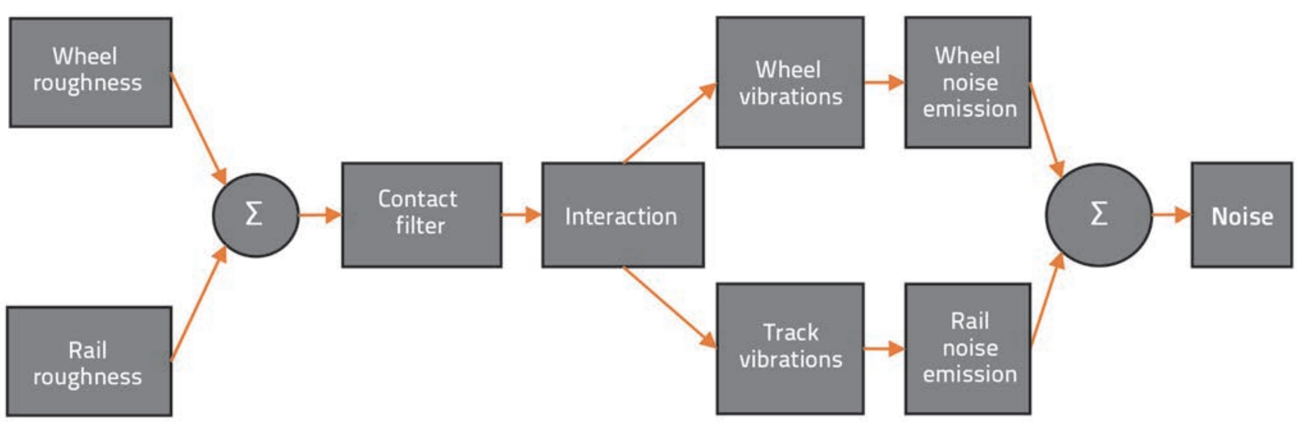

Figure 2. Rolling noise generation model [1]

that are responsible for the generation of rolling noise range from 5 to 500 mm. Vibrations are transferred to the wheel and to the track whose elements serve as noise "radiators". Track and wheel vibrations are often considered to be significant contributors to the total noise emitted to the environment. The rolling noise has a wide spectrum of frequencies, and the significance of higher frequencies increases with an increase in the vehicle operating speed.

An important property of vibrations induced by rolling activity is that these vibrations are caused by the combined roughness of the rolling surfaces of the wheels and the rails. On a typical example of a wheel with block brakes, the rolling surface has the roughness with the wavelength ranging from 40 to 80 mm. At the vehicle speed of 100 km/h, this roughness generates frequencies at which the track emits the highest level of noise. In such situation, it is difficult to determine the component that is responsible for the spreading of noise into the environment. It should be noted here that the noise caused by the wheel impacting the rail at discontinuity points, such as rail joints, flat points on wheels will not be considered although, if in fact present, they do have a significant influence on the total noise levels generated by the passing rail vehicle.

If the sound record of a five-bogie electric train (with the first and the last bogies being the traction bogies) operating on Croatian railways along the Pan-European Corridor X is considered, then the following phenomenon can be observed based on sound pressure measurements. Before the train arrives, the air pressure increases and, after the train passes, the air pressure reduces. This increase is explained by the "rail singing" due to vibrations that are generated in the rails. Maximum pressure is achieved at the passage of each bogie next to the microphone, which suggests that wheels are a significant source of noise. Bogies can be identified from the distance of 7.5 m from the track but, by bringing microphones closer to the source, even individual bogie wheels can be singled out. Running surfaces of wheels and rails are not ideally smooth and so irregularities cause displacements of wheels relative to the rail. If the wave of the wavelength λ is generated on the rail (or wheel), and if the vehicle is moving at the velocity of V, then sinusoidal vibrations are generated at the frequency of:

$$f = \frac{V}{\lambda}[\mathrm{Hz}] \tag{6}$$

The resulting high-frequency vibrations are transferred to both elements – the rail and the wheel, and their vibrations are transferred by air in form of sound waves. Wavelengths significant for noise generation can be determined from equation (6), and they cover the range from one-tenth of a centimetre, to the size of the contact surface (about one centimetre). Such irregularity of the driving surface is usually called roughness. The term roughness can be inappropriate in this context as roughness is normally used for wavelength irregularities in the range of one millimetre or less (micro-roughness). While micro-roughness is very significant for adhesion (friction when starting or braking), longer wavelengths (macro-roughness) are disadvantageous.

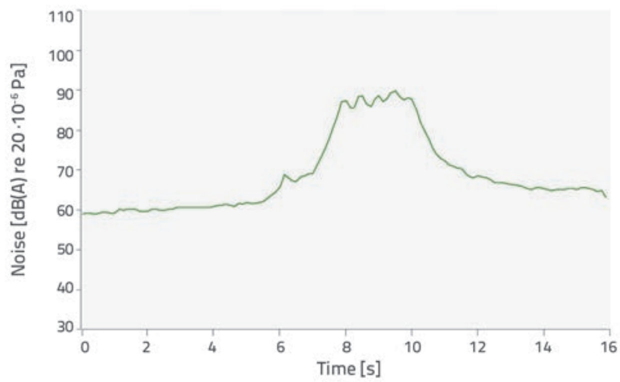

Figure 3. Time record of sound pressure for electric train passing at the speed of 160 km/h

2.1.1. Influence of driving speed and rolling surface roughness

The vehicle rolling noise increases with driving speed. The A-weighted sound pressure level is normally proportional to the speed logarithm, according to the following equation:

$$L_p = L_{p0} + N\,log_{10}\left(\frac{V}{V_0}\right) \tag{7}$$

where L_{p0} is the sound pressure at reference velocity V_0. The value of speed exponent N, defined by in situ measurement

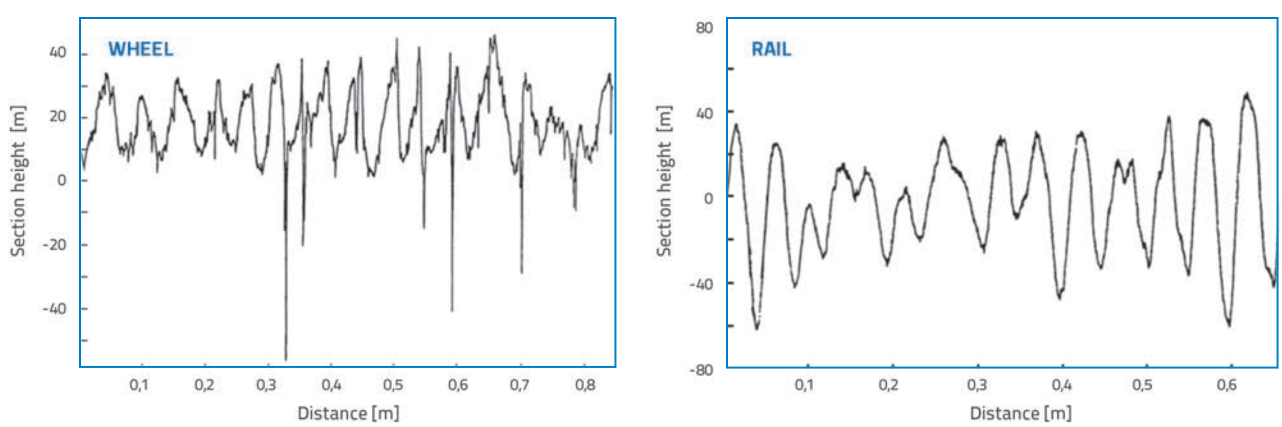

Figure 4. Section of wheel running surface with steel braking shoes (left) and section of the rail running surface with corrugation (right)

based on linear regression, mostly ranges from 25 to 35, and the typical value is 30. It can be seen from this equation that speed doubling corresponds to the 8 to 10 dB increase of the A-weighted noise level [1].

As to train wheels, they can be classified, with regard to noise emission levels, into two categories: with brake shoes on the wheel running surface, and without brake shoes (e.g., with disk brakes). The difference in noise levels between these two wheel groups varies from 8 to 10 dB, regardless of the speed. This is due to the running surface roughness of the wheel with braking shoes, which has a pronounced wavelength of about 50 mm (Figure 4). Wheels without braking shoes are much smoother. At the speed of 160 km/h (44 m/s) these irregularities result in high-frequency vibrations of 900 Hz, according to equation (6). At that frequency, the vibration displacement of 50 μm from top to top (effectively 17 μm RMS) is equal to the effective (RMS) vibration velocity of 0.1 m/s and effective (RMS) vibration acceleration of 560 m/s^2. It should be noted that composite braking shoes exert a smaller influence on wheel roughness, and are therefore more favourable compared to steel braking shoes.

If we observe the rail, it also contains periodic roughness and corrugation zones caused during the railway use, Figure 4. Rail corrugation can result in an increase of noise by 10 dB for vehicles with steel brake shoes, i.e. by as much as 20 dB for otherwise much less loud vehicles with disk brakes [1]. At rough/corrugated rails, the vehicles with disk breaks and brake shoes have similar noise levels, which prove that the excitation is defined by the sum of the wheel and rail roughness values.

2.1.2. Source of noise – rail or wheel?

In order to achieve noise reductions in any kind of system, it is of highest significance to determine the source of vibrations or, in case of several sources of vibration, what is their significance relative to each other. Otherwise, significant efforts could be made to reduce noise component of the source that contributes only slightly to the reduction of overall noise levels. That is why in case of vehicle rolling noise it is significant to determine to what extent each component (rail or wheel) contributes to the total emission of noise. It has so far been established that the influence of each of the two components is significant, although concrete results may vary from case to case.

Earlies studies of this phenomenon date back to 1984 when noise and vibrations were measured in the United Kingdom at six points on wheels, and on ten points on rails, while the running surface roughness and speed parameters (from 40 km/h to 160 km/h) were varied. The range of A-weighted sound pressure levels in excess of 40 dB was registered [7]. It is very difficult to determine by measurements alone the real proportion of noise generated by each component and so, in addition to data collection, it was also necessary to develop reliable noise propagation models for wheels and rails.

2.2. Track vibration decay rate

One of key properties for determining participation of track in the total rail traffic noise and vibration levels is the vibration decay rate. This is a vibro-acoustic property that describes how many vibrations the track can absorb, i.e. how far do vibrations travel through the rail from the excitation source before they are fully attenuated. The longer the rail section that vibrates, the greater the noise emitted during passage of the vehicle [12]. The track with a greater vibration decay rate will absorb a greater vibration energy and will transfer less vibrations to the surrounding soil and less noise to the environment, Figure 5.

Researchers started defining the track vibration attenuation problem at the phase when basic theoretical models for rail noise propagation into environment were being defined [17, 18]. In his paper published in 1997, Mr. Thompson, D.J.,a leading researcher in this area, formulates basic theoretical models and experimental measurements regarding propagation of waves through the track and their attenuation [19].

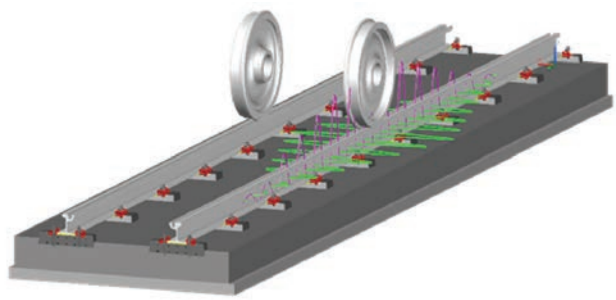

Figure 5. Attenuation of vehicle vibrations along rails in horizontal and vertical directions

The track vibration decay rate is used as a means to characterize dynamic behaviour of the track. It is expressed in vertical and horizontal directions with respect to the rail cross section. The attenuation of track vibrations, although much more significant compared to wheel attenuation, does not greatly affect the frequency response of the structure whose vibrations range from 500 to 1000 Hz, when vibrations propagate via the rails.

The attenuation of vibrations has two sources: losses of vibration energy on elastic fastening systems (setting pads, vibration attenuators, etc.) and losses of energy transferred to rail sleepers and surrounding soil, which also influence the total attenuation of track vibrations [1]. In addition, permanent way components prevent transfer of low-frequency vibrations along the rails. Although this results in a high vibration decay rate, this phenomenon is not regarded as a direct attenuation of vibrations. The rail acts as an infinite girder resting on an elastic support. This is why its decay rate is characterised by spatial attenuation of waves in the longitudinal direction. Although various forms of waves propagate through the rail (such as bending waves, longitudinal waves, and torsional waves), its oscillation can be described by the vertical (dominant) and horizontal vibration decay rates.

If velocity field on a part of a rail is composed of rigid vertical and horizontal displacements with amplitude of v_{wave}, and with the corresponding vertical and horizontal bending waves included in a simple track model, then the sound power of an individual wave of vertical or horizontal spreading can be expressed as follows:

$$W_{val} = \tilde{W}_{val} \int_{-\infty}^{\infty} \left| v_{val}\left(z\right) \right|^2 dz \tag{8}$$

where w_{wave} is the sound power per unit of length that is generated via vertical or horizontal rail oscillation speed. $V_{wave}(z)$ is the variation of vertical or horizontal wave through the rail. It is assumed that the waves passing through the rail weaken exponentially with the distance z from the place of excitation (wheel and rail contact) and where β_{wave} is the wave-specific attenuation constant. The sound power due to horizontal or vertical vibration waves passing through the rail can be described as:

$$W_{val} = \tilde{W}_{val} \int_{-\infty}^{\infty} v_{val}^2\left(0\right) e^{-2\beta_{val}|z|} dz = 2\tilde{W}_{val} v_{val}^2\left(0\right) \int_{0}^{\infty} e^{-2\beta_{val}|z|} dz = 2\tilde{W}_{val} v_{val}^2\left(0\right) \frac{1}{2\beta_{val}} \tag{9}$$

and can be replaced with the decay rate expressed in dB per meter, which is hereinafter referred to as the DR (decay rate), according to the expression:

$$DR = 20 log_{10}\left(e^{\beta_{val}}\right) \approx 8{,}686\beta_{val} \quad [\text{dB/m}] \tag{10}$$

The equation (9) shows that the vertical and horizontal decay rates directly influence the quantity of noise generated during passage of a rail vehicle [20]. A two-time increase in the vibration decay rate results in the 3 dB decrease of noise emitted by the rail into the environment [21].

An idealized view of the exponential vibration decay rate at the distance z from the vibration excitation point is given in Figure 6. Actual situations greatly deviate from this basic model due to a variety of factors that influence propagation of vibrations along the rail.

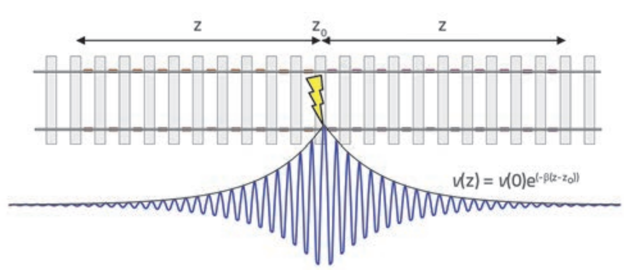

Figure 6. Decay of vibrations as related to distance from excitation point [22]

The decay of vibrations at the rail to sleeper contact is greatly influenced by the under rail pad thickness, Figure 7. A detailed elaboration of the under rail pad influence on the track vibrations decay rate, with comparison of models and in situ measurements, is presented in the scope of the project prepared by researcher Graupeter, T. mentored by Jones J.C.J., as related to the dependence of frequencies and under rail pad stiffness values during calculation of the vibration decay rate [23]. The amplitude of vibrations reduces across the rail quasi-exponentially with the distance along the rail. The better the decay properties, the faster the reduction of vibrations. The parameter that is used to describe the decay rate is expressed in dB/m, and is mostly presented on a logarithmic scale.

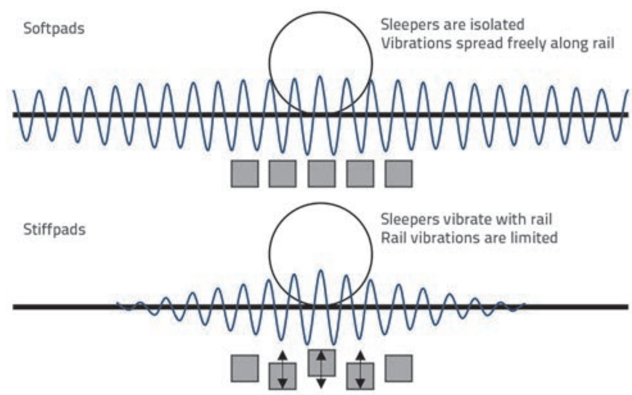

Figure 7. Influence of setting pad stiffness on the sleeper to rail connection, and on vibration decay across rail [1]

The track vibration decay rate has numerous applications. Thus it can be used:
- to determine dynamic properties of tracks for typical noise measurements (for ISO 3095 [3]), i.e. to determine track suitability for the conduct of typical measurements,
- as dynamic parameter in the separation of noise sources (vehicle and track) [24],
- to determine indirect roughness of rail vehicle wheels [24],
- as parameter for evaluation of various vibration attenuation systems [21, 25].

The experience gained so far in the numerical modelling of dynamic properties of tracks shows that the most significant parameters are: rail geometry and material, static and dynamic stiffness of under rail pads and decay factor, type of fastening, geometry, material and distance between the sleepers. If all these data are known, then the decay rate can be determined by means of theoretical models such as TWINS [13]. As the vibration decay rate depends on a large number of parameters that vary along real-life railway sections depending on the route geometry, age, and maintenance, it is recommended to conduct direct decay rate measurements by means of in situ measurements.

A parameter can be determined experimentally by measuring frequency response functions (FRF) using the modal hammer as a source of excitation along the rail and accelerometers for measuring vibrations [20]. The decay rate can also be measured by accelerometers in relation to the passage of rail vehicles [26]. Frequency response functions are then averaged into frequency bands (mostly one-thirdbands) from which vibration decay is defined, depending on the distance from the excitation point, either via approximation curves (by straight-line slope estimation) or by analytical procedures. Experimental methods will be additionally explained and analysed later in this paper.

3. Experimental research

The experimental part of the research is the phase in which relevant field measurement data about vibration decay rate are collected. The data were collected on the previously selected standard rail sections by direct in situ measurements. The data were collected in accordance with requirements for the selection of teste sections, according to [3, 5].

Figure 8. Double track rail line Vrpolje – Ivankovo (position: KM 183+900) (left), and the rail installed at the same section (right)

3.1. Description of test sections

3.1.1. Vrpolje – Ivankovo section

The test site was formed at KM 183+900 of the Vrpolje – Ivankovo Section of the double track international railway line M105 Zagreb TS – Tovarnik – NB (national border). This test section is situated at a highly accessible location. Here the train speeds attain 160 km/h and so typical noise measurements are made in this zone for newly designed trains [27]. The Vrpolje – Ivankovo rail section is a part of the HR1. TEN-T Corridor (Pan-European Corridor X). It is a double track line with the 4.0 m axis-to-axis distance between tracks. The railway structure is of traditional type, i.e. the permanent way consists of: rails, reinforced-concrete sleepers, elastic fastening devices, and ballast prism, Figure 8.

Rails

Rails of type 60 E1 (according to EN 13674 – 1) or UIC 60 (according to UIC Code 860) were installed at this railway section. These rails bear the mark: ZENICA 89 IV UIC 60П——, as shown in Figure 8. Based on this mark, it can be concluded that these rails are wear-resistant (minimum tensile strength: 880 N/mm2, hardness: 260 to 300 HB), and that they were manufactured in April 1989 at Zenica Steel Works.

Rail fastening system

The spring clip type SKL-8 with the corresponding steel rib leaning onto the concrete sleeper, and with an insulator between the steel rib and the concrete sleeper, Figure 9, was used for connecting rails with concrete sleepers at the Vrpolje – Ivankovo railway section. This clip was made according to the licence granted by company VOSSLOH (which is now rarely applied), and is still in use at tracks where it was installed during renovation, or at new railways built in the period from 1980 to 1990.

Sleepers

One-piece transverse reinforced-concrete sleepers were installed at the studied Vrpolje – Ivankovo railway section. After inspection of the sleepers, it was established that sleepers type PB85 manufactured in 1989 byGraditelj, and sleepers type PB 85K, manufactured in 2005 by Vibrobeton, Vinkovci, were used on this section, Figure 9.

Ballast

As can be seen in the preceding figures, traditional ballast bed was constructed at the Vrpolje – Ivankovo railway section. Track

Figure 9. Rail fastened with elastic clip SKL-8 (left side), reinforced concrete sleepers at test section (centre) and ballast prism (right side)

inspection revealed that the ballast bed is properly filled, which certainly contributes to lateral stability of the track. The ballast bed was made on the embankment ranging from 1 to 2 m in height (Figure 9) and, according to information from design documents, a protective (subbase) course, 40 cm in thickness, was constructed between the embankment and the ballast bed.

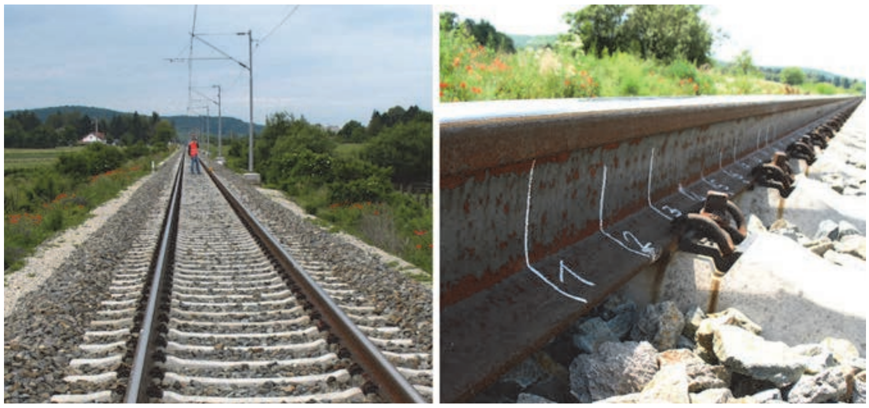

Figure 10. Koprivnica – Križevcisingle track railway line (left) and installed rail 60E1 (right)

3.1.2. Koprivnica – Križevci Section

The second test section is situated on the railway line M201 (NB – Botovo – Dugo Selo), between Koprivnica and Križevci, on the Pan-European Corridor Vc. The line was rehabilitated in 2010, just before the start of the vibration decay rate testing [28]. This section is also located in a highly accessible zone with travelling speeds of up to 160 km/h. The Koprivnica – Križevci railway line was realized as a single track line, and the test section lies in straight line. The track structure (permanent way) is composed of: rails (60E1), reinforced-concrete sleepers, elastic clip (SKL-14), and ballast bed, Figure 10.

Rails
Rails type 60 E1 (according to EN 13674 – 1) or UIC 60 (according to UIC Code 860) were installed at this rail section, Figure 11.

Rail fastening accessories
The spring clip type SKL-14 with the corresponding steel rib leaning onto the concrete sleeper, Figure 11, was used for connecting rails with concrete sleepers at the Koprivnica – Križevci railway section.

Sleepers
One-piece transverse reinforced-concrete sleepers were installed at the studied Koprivnica - Križevci railway section. After inspection of the sleepers, it was established that the sleepers type PB85K manufactured by Vibrobeton, Vinkovci, were used on this section, Figure 9.

Ballast
Traditional ballast bed was realized at the Koprivnica – Križevci rail section. The track inspection revealed that the ballast bed is properly filled.

3.2. Data collection

The vibration decay rate measurements are regulated in the scope of HRN EN 15461:2011. The equipment used fully complies with measurement requirements. The decay rate measurements can be conducted in situ on the track section under study using the modal hammer and accelerometers. The measurements are made to determine vertical and horizontal waves as a function of distance based on frequency response functions for a specified disposition of measurement points along the rail.
The test section must meet some basic requirements so that measurements can properly be conducted. These requirements imply invariability of those track parameters that may influence decay rate measurements along the test section, such as the rail cross section, stiffness of under rail pads, track superelevation, and axis-to-axis distance between supports (sleepers) in case of a discretely supported rail. Rails must also be welded continuously, without expansion joints.
Two measurement approaches are possible: the accelerometer can be positioned at a fixed point and the hammer (as the

Figure 11. Single track elastic clip SKL-14 (left), sleepers type HŽ – PB85-K (centre) and ballast prism (right)

source of excitation) can be moved along measurement points, or the hammer can be used to apply impacts at a fixed point on the rail while accelerometer is moved along measurement points. The first solution – selected for the purposes of this paper – has proven to be much more practical in most cases, although it should be noted that measurement results are identical, regardless of the approach used. The approach involving a fixed accelerometer position is presented in the standard for measuring dynamic response of track structures [5].

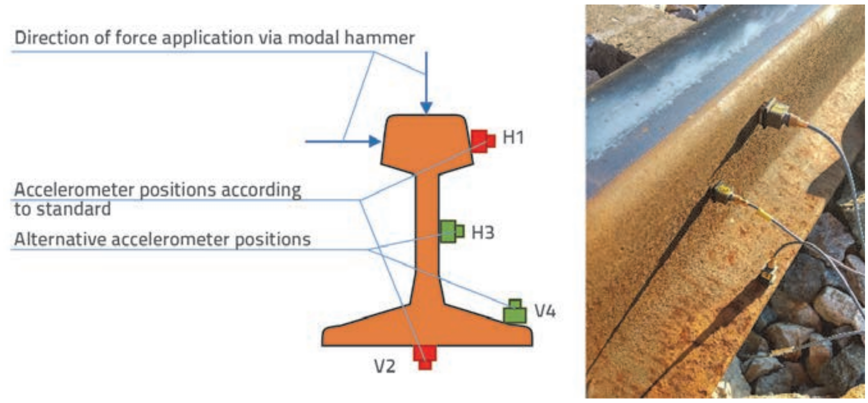

Figure 12. Accelerometer positions along rail cross section for decay rate measurements and direction of forces in horizontal and vertical direction

The accelerometer placed at a fixed position along the test section must meet the following requirements:

- it must be situated at the half span between two discrete supports (sleepers);
- it must be more than 3 m away from the pumping sleepers (sleepers not adequately positioned in the ballast material);
- fastening accessories on sleepers in the immediate vicinity of accelerometers must not be damaged or poorly fastened;
- it must be more than 5 m away from the rail weld;
- it must be at least 40 m away from the expansion joint on the track, if such joint is present.

The accelerometers are attached to the rail in the vertical and horizontal direction with respect to the rail cross section. In the vertical direction the accelerometer must be positioned at the centre of the rail head or at the centre of the rail foot (if the accelerometer can not be positioned on the rail head, which is often the case when measurements are made under traffic). In the horizontal direction, the accelerometer must be placed at the external edge of the rail head, Figure 12 (accelerometers marked in red). In addition to the positioning required by the standard, the accelerometers were also placed as shown in Figure 12 (accelerometers marked in green). These accelerometer positions were selected so as to compare results for various accelerometer positions for the purpose of future measurements on track structures with a specific closing system until the running surface of the rail, where it is often impossible to place accelerometers in the position specified in the standard. Such structures are often encountered in urban areas where road and rail vehicle surfaces intersect (metro, light urban railway, tram).

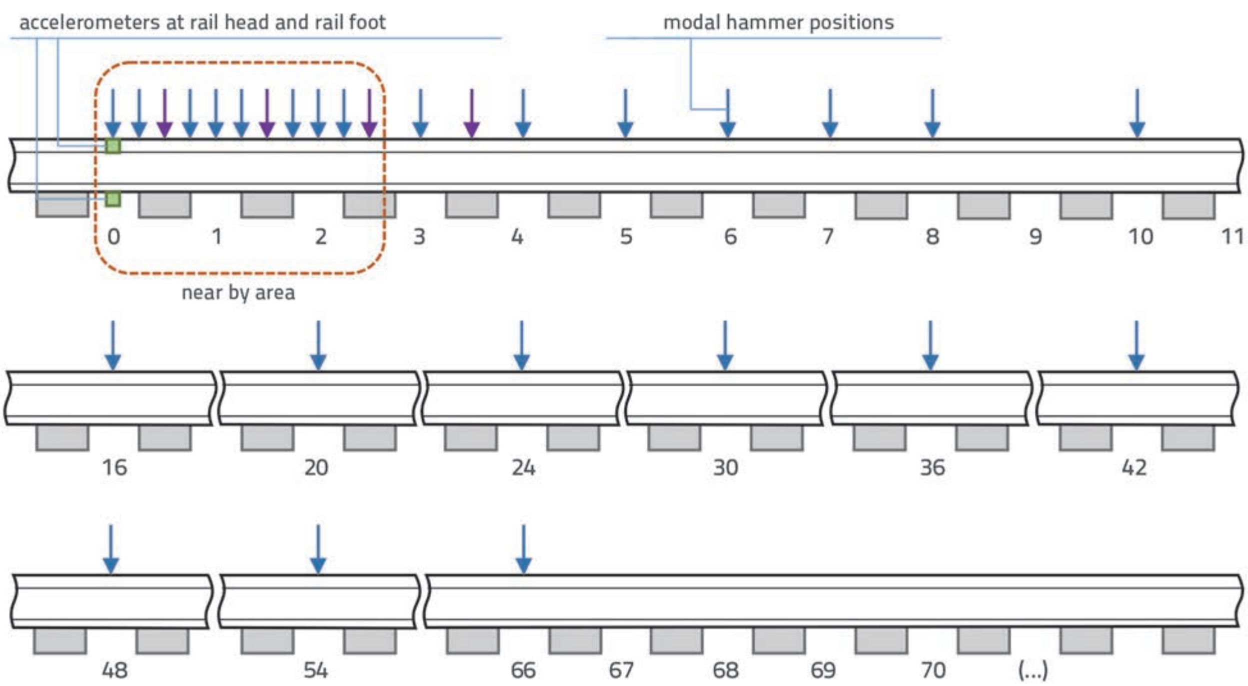

Figure 13. Disposition of measurement points along rail as defined by NOEMIE project and accepted by standard [5]

Accelerometers are used to measure acceleration or (if analogous integration exists) displacement of rails. It is significant to conduct measurements on a sufficient number of measurement points along the rail, in accordance with a predefined disposition of points. The number of measurement points and their disposition has been studied and defined in the scope of NOEMIE project [29], and has been later accepted by standard [5], Figure 13. This disposition of measurement points has later been criticized in research conducted in the scope of the STARDAMP project [22] where it is indicated that a nearby area must be expanded to a minimum of three sleeper spaces, so that a more accurate characterisation of decay rate can be achieved. Spreading out of measurement points aimed at simplifying the measurements, while keeping the same accuracy of results, is considered in the scope of the QUIET-TRACK project [30]. Dense disposition of points is necessary to enable reliable determination of decay rate in case of low frequencies of vibration where an average decay rate is 10 dB/m. The positioning of hammer further away from the initial position becomes important in case of lower decay rates that occur at higher frequencies.

For more distant points, vibrations need not be measured in every zone between the sleepers, and so the measurement point network becomes less dense as the distance increases. At discretely supported track structures, equivalent hammer positions must be ensured due to variation in response between the supports in the vicinity and above the first resonant frequency of rail resting on sleepers spaced at equal 0.6 m intervals.

The frequency response function for rail vibrations resulting from modal hammer impacts is measured in vertical direction via accelerometers along the running surface or foot of the rail. In horizontal direction, this function is measured via accelerometers placed at the running edge of the rail head. The functions are presented in one-third bands of the spectrum, in the minimum 100 Hz to 5000 Hz range. During the frequency response function measurements, the data are averaged at every measurement position for no less than four consecutive hammer impacts and, at that, care must also be taken about the coherence of measurement results for each measured position. After collection of field data, results were analysed and interpreted using two methods – determination of decay rate by direct procedure via inclination of decay line, or by analytical calculation as proposed in standard [5].

3.3. Analysis of data

To enable comparison of measurement results, the data collected were analysed in two ways, as shown further on in Section 3.3. In addition, the results obtained were subsequently compared to check accuracy of the simplified method for determining the decay rate based on decay line slope. The proposed method requires less time for analytical calculation during the decay rate determination.

3.3.1. Determination of decay rate by standard method

The determination of the vibration decay rate through direct analytical approach by summing frequency responses shown in equation (9) has proven to be quite reliable in practical applications. If *A* is the acceleration function of vibration velocity, then we have

$$\frac{1}{2\beta}=\int_0^\infty \frac{\left|A(z_i)\right|^2}{\left|A(0)\right|^2}dz \approx \sum_{z_i=0}^{z_{max}} \frac{\left|A(z_i)\right|^2}{\left|A(0)\right|^2}\Delta z_i \tag{11}$$

where z_{max} is the most remote measurement point, and all measurement locations are summed up.Just like in case of integral determination using the rule of average interval values where intervals are not constant, Δz_i represents, for each location in the sum, the distance from central points in the interval to the end of interval from each side. The influence of the interval should be the smallest at the measurement point z_{max} but it must be symmetrical around z_{max}, while Δz_0 must be calculated from z = 0. Thus we have

$$DR(f_c)\approx\frac{4.343\left|A(0)\right|^2}{\sum_{z_i=0}^{z_{max}}\left|A(z_i)\right|^2\Delta z_i} \tag{12}$$

DR is determined in the frequency range for individual one-third bands around the central frequency f_c, usually in the range from 100 Hz to 5000 Hz. It can be noticed that the measurement *A(0)* is extremely important as it occurs as a constant factor in the equation. That is why it is very important to accurately determine *A(0)*during measurements, as specially emphasized in paper [22]. It can be argued that this is one of deficiencies of analytical approach in the decay rate measurement, which is not so significant in the straight line – slope estimation method described hereunder. If A(0) is accurately defined, the sum of frequency responses according to (12) is, unlike the line-slope approximation method, much more robust as it places emphasis on greater frequency response values in which there are less disturbances as compared to other types of waves present in the rail. The decay rate can erroneously be determined if the practical value z_{max} is lowered in any one-third band, before a sufficient decay occurs. The minimum decay rate can be expressed for a specific value of z_{max} from (13) under assumption that no decay is exhibited by function A, i.e. that $A(z_i)$ is equal to *A(0)*:

$$DR_{min}(f_c)=\frac{4{,}343}{z_{max}} \tag{13}$$

The analytically obtained decay rate should be compared with this value and, if it is lower than or close to this value, the decay rate estimation can be considered unreliable [20].

The method described in this way was published in 2011 as European standard for measuring dynamic properties of railway tracks. It is entitled HRN EN 15461:2011: *Railway applications. Noise emission. Characterization of the dynamic properties of track selections for pass by noise measurements* (HRN EN 15461:2008+A1:2010) [5]. The first edition of this standard was published in 2008, and the standard was extended in 2011. This standard is presented in the scope of Technical specifications for interoperability – TSI Noise, the last version of which was issued in 2014 [2]. The noise measurement for new and renovated rail vehicles is regulated by technical specifications contained in HRN EN ISO 3095:2013: *Acoustics – Railway Applications – Measurement of noise emitted by rail bound vehicles* (*ISO 3095:2013; EN ISO 3095:2013*) [3], which refers - for selection of test sections - to the above mentioned standard [5] and to the standard related to the roughness of running surfaces of rails [4].

The standard *HRN EN ISO 3095:2013* [3] also determines the bottom limit curve for the horizontal and vertical decay rate, which is to be met by tracks to enable conduct of typical measurements of noise generated by the passage of rail vehicles, Figure 14.

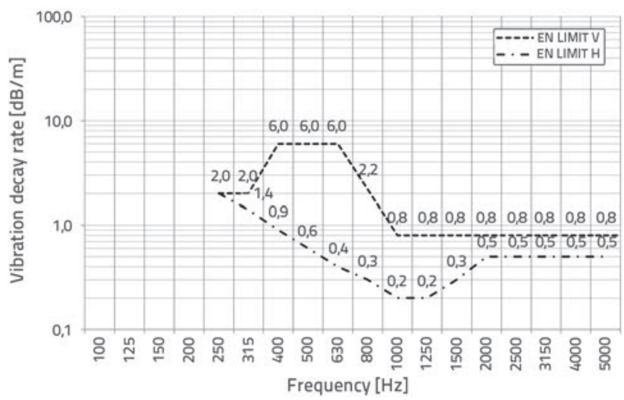

Figure14. Bottom limit decay-rate curve for typical noise measurements according to [3]

3.3.2. Determination of track decay rate based slope estimation

One of the methods for estimating the vibration decay rate is to determine the inclination from the frequency response diagram, dB, depending on the distance from the accelerometer position, for individual one-third frequency bands (from 100 Hz to 5000 Hz). This method has been proposed in research conducted by Thompson D.J. Hemsworth, B. and Vincent, N. [18, 31]. The frequency response functions in the frequency range for the one-third span around the central frequencies of 250 Hz are presented in Figure 15.

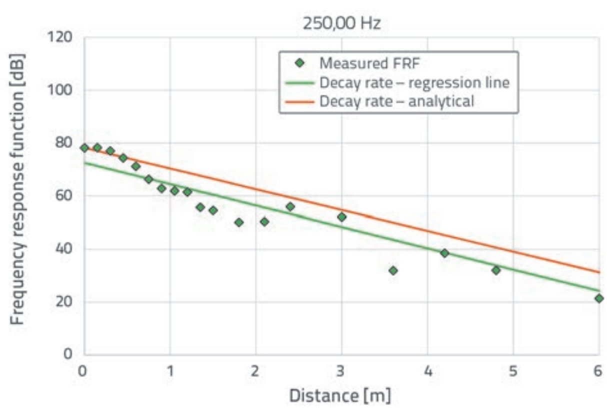

Figure 15. Frequency response function for one-third band around 250 Hz obtained during decay rate testing for standard track structure on ballast

Frequency responses are measured on the track structure with concrete sleepers, placed in ballast. The response values are presented for measurement points situated up to 6 m from the accelerometer point (0 m). It can be seen from inclination of the regression line that 250 Hz vibrations are lowered by approximately 46 dB at the distance of 6 m, which corresponds to the vibration decay rate of 7.7 dB/m. It can be seen that, in this case, the analytically obtained decay rate (7.81 dB/m) is very similar to the inclination obtained by regression of measurement points.

This approach requires an experienced test supervisor because the process of straight-line association by regression cannot be automated. The problem in the analysis of results lies in the fact that vibrations generated during excitation by modal hammer do not occur solely in horizontal and vertical directions but, in reality, there are waves in the adjoining area described by simple track model, composite action of horizontal and vertical waves, torsional waves and cross-sectional oscillation mode waves [20]. In an ideal case involving exponential reduction of vibrations (Figure 6), disturbances are still experienced due to periodic action of track supports and variation of response through the field between the supports, especially around the first two resonant frequencies of rail resting on sleepers spaced at 0.6 m intervals, which usually amounts to 950 Hz and 2200 Hz [32].

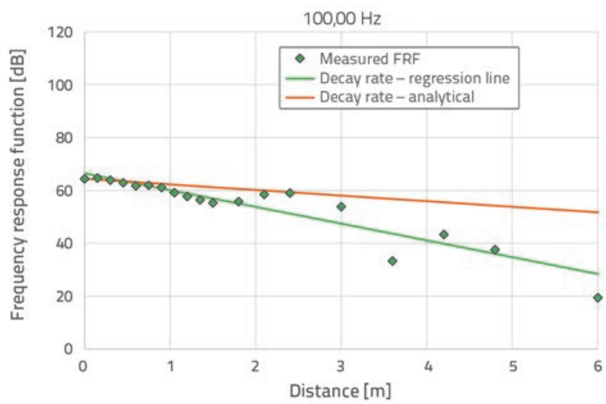

Figure16. Frequency response function for one-third band around 100 Hz obtained during decay rate testing for standard track structure on ballast

Testing site measurement results for the one-third band around the central frequency of 100 Hz are presented in Figure 16. It can be concluded from the slope of the curve that the decay rate is greater than that obtained by analytical calculation. In fact, it should be noted, especially at lower frequencies, that distant locations do not provide very reliable results because of the above mentioned occurrences, and so a greater significance should be attributed to response values that are closer to the accelerometer. Thus the analytical decay rate of 2.12 dB/h can be observed, while it can be concluded, via the slope of the curve by regression of measurement points up to 6 m, that the decay rate amounts to 6.34 dB/h. The analytical approach of frequency response determination was introduced as a reaction to such deficiencies in the manual determination of decay rate.

Table 3. Naming individual decay rate measurements

Test section	Measurement No.	Data processing method	Accelerometer position	Rail	Hammer type
KŽK_VRP-IVA	01	MC_RE	H1	STR	TČ
KŽK_KOP-KRIŽ	02	MC_NA	V2	JTR	MČ
	...		H3		
			V4		

3.4. Analysis of test results

The procedure for the analysis of data collected on testing sites is described in this Section. The data analysis procedures differ in the approach to data processing (automated analytical method and manual straight line - slope estimation method), and in the source of input data (data collected by measuring decay rate using impact hammer, and data collected at the passage of rail vehicles).

Decay rate measurements were conducted at the total of six measurement positions at testing sites along the Vrpolje – Ivankovo and Koprivnica – Križevci railway lines. At the Koprivnica – Križevici section, measurements were made at the north-side rail (STR) with a softer hammer (8208), while a harder hammer (8206) was used at the south-side rail (JTR). Vibrations were collected from 2 accelerometers for each rail, in the H1 and V2 directions. A total of 8 decay rate results were obtained on this section by the analysis of results based on two above described methods.

At the Vrpolje – Ivankovo section, measurements were made with a harder hammer and a softer hammer at four positions, while vibrations were registered with four accelerometers for each measurement position in the directions H1, V2, H3, and V4, Figure 12. A total of 32 decay rate results were obtained by the analysis of results based on two above described methods.

The sections were named (designated) based on the following characteristics: abbreviated name of the section, test number, data analysis method, accelerometer direction, rail orientation, and hammer type, which finally results in the following form of the name/designation: KŽK_VRP-IVA-01-MC_RE-H1-JTR-TČ. If the rail and hammer position is left out, then this is an average value of all measurements made according to the mentioned method on a single measurement position (which includes measurement with both hammers). Then the measurement record could read as follows: KŽK_VRP-IVA-01-MC_RE-H1.

3.4.1. Influence of type of modal hammer as the source of excitation

The impact hammer (Figure 17) with an adequately hardened impact surface is used to apply the vibration generating force. The impact hammer and accelerometers are connected to the data collection and analysis unit so as ensure good quality of measurements, and to achieve the required frequency. A smaller modal hammer with harder tip should be used for higher frequencies, while a softer-tip hammer should be used for excitation of lower vibrations [9].

Figure 17. Impact hammers – 8208 with hard plastic tip and 8206 with aluminium tip

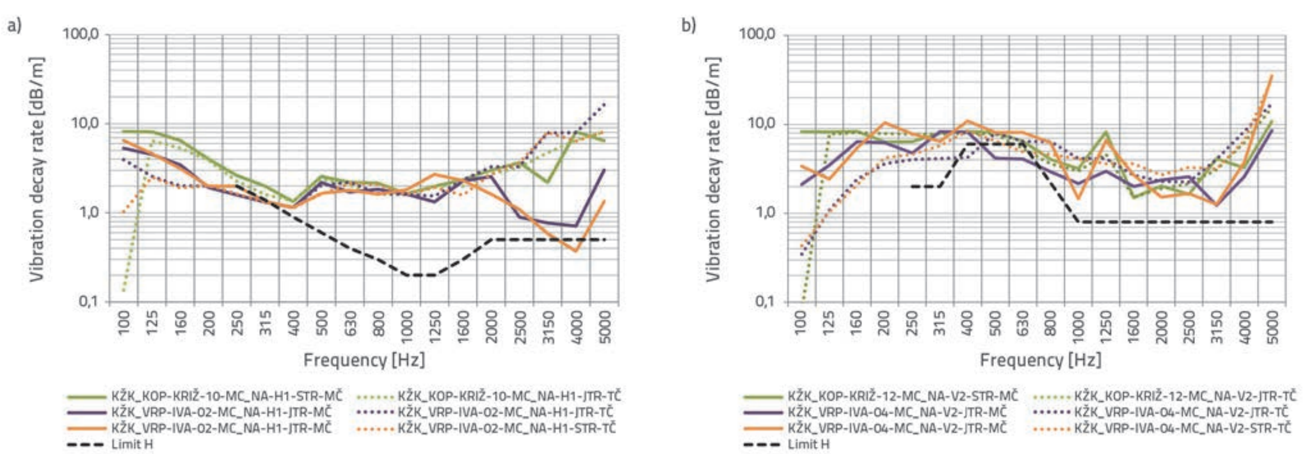

Figure 18. Comparison of hammers 8208 and 8206 as sources of excitation of vibration on sections under study, expressed via decay rate in horizontal (up) and vertical direction (down)

In the scope of these measurements, it was first of all necessary to determine which hammer can produce vibrations in the frequency range of interest. By conducting the decay rate measurements, it was established that the hammer Brüel&Kjær 8208 with a hard plastic tip is sufficient for the testing. The tests were also conducted using a harder hammer type Brüel&Kjær 8206 with aluminium tip, but comparative tests conducted on both testing sites revealed that this hammer is unlikely to produce low-frequency vibrations, Figure 18.

It can be seen from the diagram that in case of the harder-tip hammer it is not possible to sufficiently excite the rail in vertical direction at low frequencies (from 100 Hz to 250 Hz), and that the decay rate is lower than that obtained with the softer-tip hammer. An average frequency range (from 315 Hz to 2000 Hz) is equal in both directions, regardless of the hammer used, while in horizontal direction different results are obtained by harder and softer hammer at higher frequencies (2500 Hz to 4000 Hz), where the softer source of excitation (hammer 8028) provides somewhat lower decay rate results.

3.4.2. Comparison of data processing methods

The decay rate was determined at test sections on six positions based on two distinct data processing methods. The comparison of these data processing methods is given below. Vibration decay rate results are taken as average values of measurements conducted by harder and softer hammers, at accelerometer positions H1 and V2.

The results show that none of the two methods exhibits significant deviations. It can however be seen that standard deviation is somewhat higher at both sections in horizontal direction, Table 4.

Table 4. Standard deviation of methods MC_NA and MC_RE on test sections

Section	Standard deviation (verticaldirection) [dB/m]	Standard deviation (horizontal direction) [dB/m]
KŽK_VRP-IVA	1,08	1,35
KŽK_KOP-KRIŽ	1,13	1,30

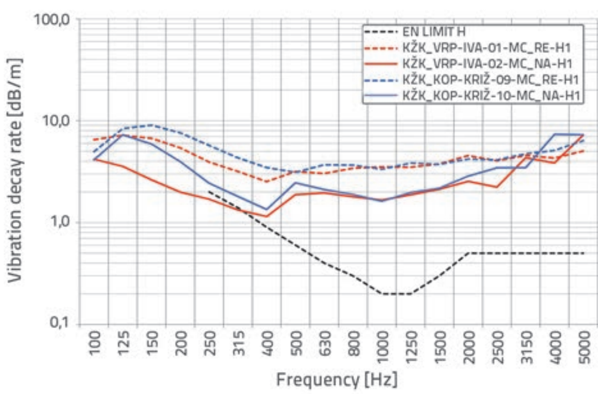

Figure 19. Comparison of methods for determining decay rate in horizontal direction

It can be concluded from these results that the decay rate is overestimated by the MC_RE method as compared to MC_NA method, Figure 19 and Figure 20.

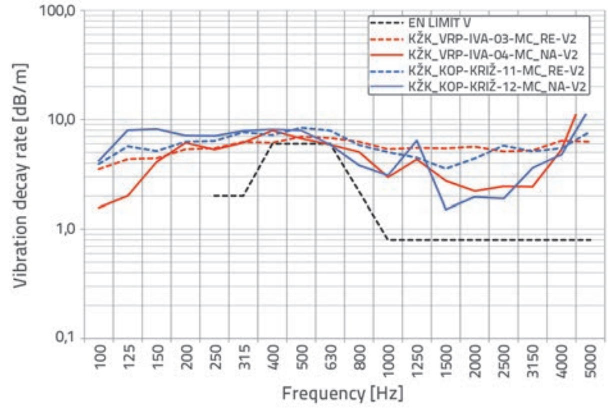

Figure 20. Comparison of methods for determining decay rate in vertical direction

3.4.3. Influence of accelerometer position on test results

The decay rate on sections under study was determined from the rail acceleration time signal via accelerometers attached to the rail in horizontal or vertical directions as related to its cross section.

At the section KŽK_VRP-IVA, with the completely free access to rail, the comparison was made for accelerometer measurement positions H1 and H3 in horizontal direction, and for V2 and V4 in vertical direction. Positions H1 and V2 were arranged in accordance with standard [5], Figure 12.

At the test section KŽK_VRP-IVA, the decay rate was measured at the north and south rails of the south-side track, using hard and soft hammers as the source of excitation, at measurement positions H1, V2, H3 and V4, which provides a very good insight on the influence of the measurement position on the measured decay rate, Figure 21 and Figure 22.

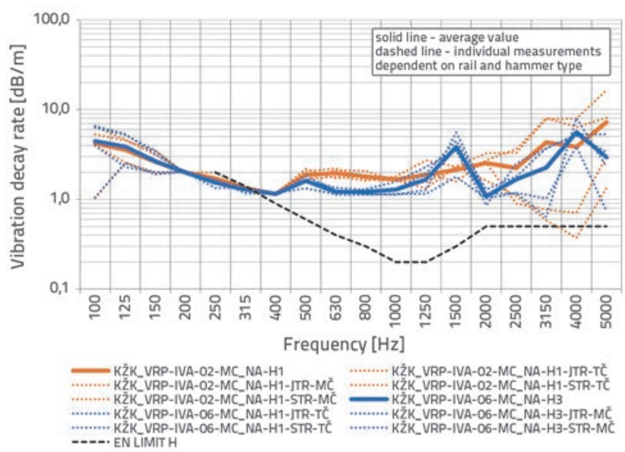

Figure 21. Comparison of decay rate in horizontal direction at two accelerometer positions (H1 and H3

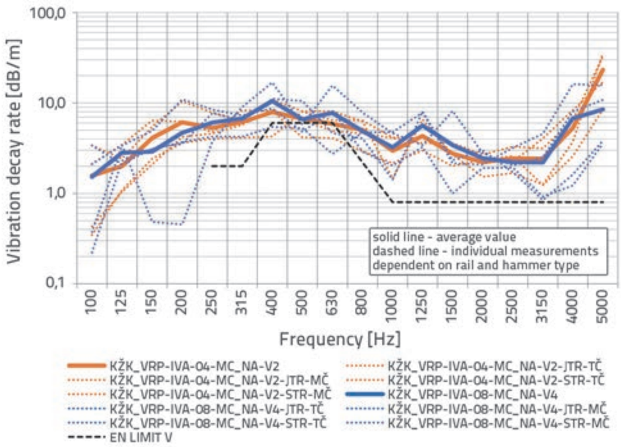

Figure 22. Comparison of decay rate in vertical direction at two accelerometer positions (V2 and V4)

These diagrams show decay rate measurements using hard and soft hammers, as well as the average value calculation for accelerometer positions H1 and H3 for horizontal direction, and V2 and V4 for vertical direction.

The absolute difference between average decay rates for accelerometer pairs H1-H3 is also presented to facilitate understanding. This enables determination of the difference in response rate, Figure 23 and Figure 24.

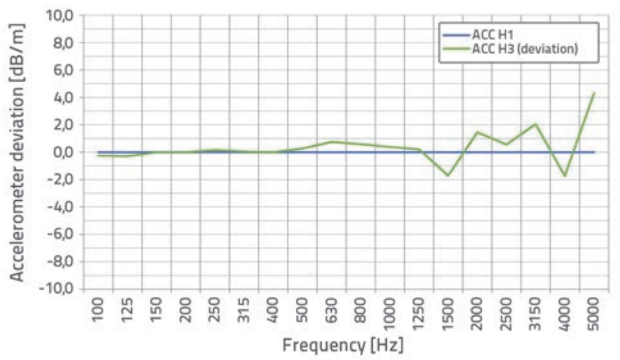

Figure 23. Average deviation of accelerometer H3 from accelerometer H1

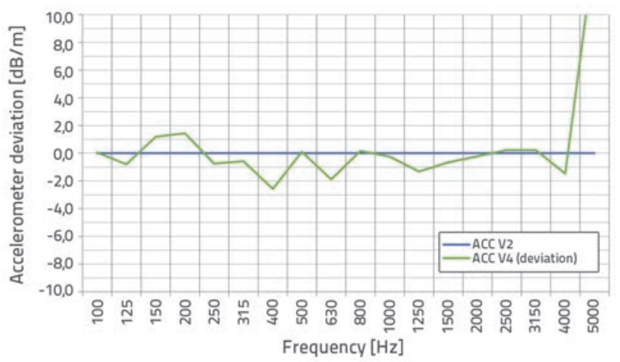

Figure 24. Average deviation of accelerometer V4 from accelerometer V2

It can be seen that the greatest accelerometer deviation in horizontal and vertical directions is situated around the central frequency of 5000 Hz, while the correspondence of the remaining results is very good. The standard deviation of accelerometer H3 as related to accelerometer H1 amounts to 0.44 dB/m, while the standard deviation of accelerometer V4 as related to accelerometer V2 amounts to 0.58 dB/m.

4. Conclusion

The conduct of decay rate measurements on railway tracks is a highly complex engineering task. Before the testing, it is necessary to properly determine the measurement equipment settings, and to study the disposition of measurement positions and the conditions that the track must meet to be eligible for this type of testing.

The vibration decay rate measurements by modal hammer, as described in standard [5], require a highly accurate determination of the frequency response function at the position x = 0, Figure 13, as the said value is the reference value for all further positions on which response caused by modal hammer is measured.

The influence of hammer tip hardness on decay rate results was also studied. It was concluded that the harder aluminium-made hammer tip cannot adequately generate vibrations in a lower frequency range (from 100 Hz to 250 Hz), while the softer hammer tip causes somewhat greater dissipation of results at higher frequencies (4000 Hz to 5000 Hz) in horizontal direction.

A good correspondence of results was obtained for the two data processing methods used: analytic method and straight line - slope estimation. However, decay values are slightly overestimated in the line – slope determination. It is therefore advisable to use the analytic method specified in the standard, and to use the straight line – slope method to check measurement results so as to eliminate errors that cannot be observed using the analytic procedure.

An alternative accelerometer position at the rail cross-section was also considered at the testing site Vrpolje – Ivankovo for the case when measurement positions defined in the standard were not accessible (in case of closed track structures that can only be accessed via shafts along the external edge of the rail). The deviation from standard accelerometer positions in vertical and horizontal directions amounted to 0.58 dB/m and 0.44 dB/m, respectively. It can therefore be concluded that decay rate measurements can be made at alternative accelerometer positions (H3 and V4) in exceptional cases when measurement positions specified in the standard are not accessible.

It should finally be noted that both test sections comply with the vibration decay values specified for vertical direction, while the test section Vrpolje – Ivankovo exhibits slightly lower results in the area around central frequencies (250 Hz to 315 Hz) where, according to standard method, it does not met the vibration decay rate requirements for horizontal direction.

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