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ASSESSING THE FREQUENCY-DEPENDENT RAILPAD STIFFNESS AND DAMPING FOR COMBINED VIBRATION AND NOISE REDUCTION

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ABSTRACT

Railway noise at moderate speeds is determined by the wheel-rail contact. The combined roughness leads to a dynamic contact force which is defined by the mobility of both structures. Railpads are track components that strongly affect the track dynamics in the important audible range, and low-frequency vibration transmission. It is known that hard pads lead to lower airborne noise but higher vibration transmission to the ballast bed and subgrade. Soft pads exert less strain on the ballast which is advantageous for track maintenance, but inevitably lead to higher noise. A new generation of pads uses rubber with high damping, molded in complex shapes. Their visco-elastic properties and near-incompressibility yield a pad stiffness that varies strongly with frequency, and which cannot be assessed by standardized tests. We present a method to assess the pad stiffness experimentally. The pad is clamped on a sleeper under a top mass, which is excited by a vertical or lateral harmonic force. The measured acceleration of the loading mass and its excitation force yield the required frequency-dependent stiffness and damping values. These equivalent springs allow an informed choice of the ideal component given a certain track superstructure.

Keywords: *railway noise, railway vibrations, visco-elasticity*

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1. INTRODUCTION

The Swiss railway network is one of the densest in the world. In urban areas, dwellings are being constructed increasingly close to busy railway lines, thereby introducing the need for noise reduction. Railpads are the superstructure component with the highest effect on noise radiation, and they can be much easier replaced than rails or sleepers in case corrective measures are required [1]. The standard hard ethyl-vinyl-acetate (EVA) railpads used by SBB offer the lowest possible noise generation, but the stiff connection to the sleepers results in a faster ballast deterioration. An ideal railpad combines low noise radiation with better vibration mitigation, typically achieved by softer railpads.

Several solutions to achieve these opposing goals are under investigation, most of them exploring the effect of visco-elastic vibration-damping materials. In-situ testing of new designs is costly, and can only be done for a limited number of prototypes. On the other hand, lab tests only offer limited predictive accuracy of noise generation and, more generally, important dynamic properties such as the track's point mobility and track decay rate (TDR) [2]. To investigate the potential of the material properties on the track dynamics, we developed an analytical model that takes all superstructure components into account [3]. It is possible to change the properties of ballast and rail, and the frequency response function of the sleepers. The railpads are represented as three springs: one vertical, one lateral, and one rotational. We have shown that the model captures most features of measured data correctly, even in narrow-band spectra, revealing the effect of pin-pin modes and sleeper resonances. However, the model can only be predictive if the input data is accurate. Where most superstructure components are well known, railpads have a more complex behavior. Their frequency-dependent

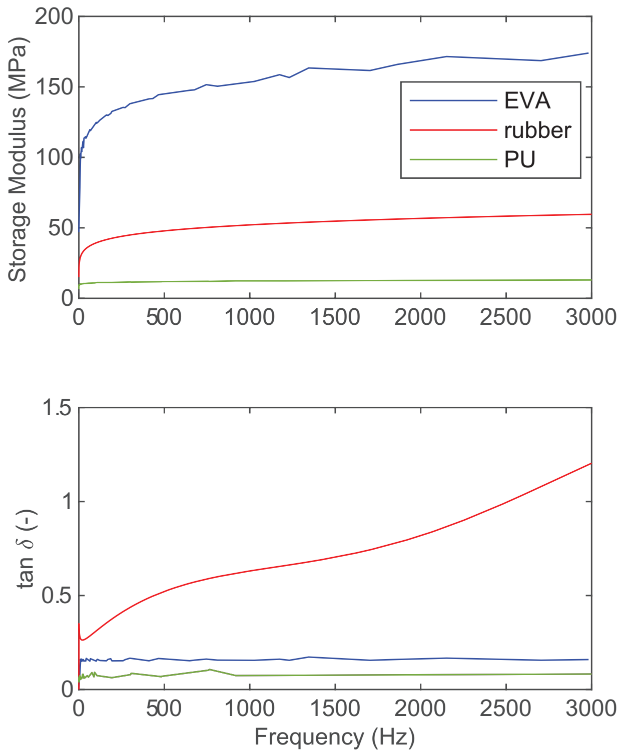

Figure 1. Storage modulus and tan δ of EVA, PU, and rubber, showing a clear difference in stiffness and damping properties.

Young's modulus and damping, and the extremely high Poisson's ratio of rubbery materials, depend on the manufacturers. Their measurement, typically done by dynamical mechanical analysis, is prone to large errors so that the calculation of equivalent springs is very uncertain. It is known that the vertical spring stiffness can vary largely over the frequency range of use [4]: from 0 Hz (static stiffness) over 10-100 Hz (ground vibrations) to 5000 Hz (noise generation). However, standards for railpads [5] only foresee measurements up to 20 Hz, which makes extrapolation to higher frequencies error sensitive.

We present an approach to measure the complex-valued and frequency-dependent spring stiffness of railpads in a lab environment. A pad is clamped under realistic conditions and is then excited by a dynamic force. In the following, we first introduce the vertical dynamic stiffness of a railpad and its effect on track decay rate (TDR). Then, we present the experimental setup, and a fitting procedure to extract the real and imaginary parts of the vertical stiffness.

2. DYNAMIC PROPERTIES OF RAILPAD MATERIALS

The Swiss railways use stiff railpads made of EVA. With a nominal Young's modulus of 20-80 MPa, and a vertical static stiffness of 800 kN/mm, the pads are extremely stiff. They are the best option for low noise generation, especially in densely populated areas, but transmit most of the vibrational energy to the ballast. In rare cases, soft polyurethane (PU) pads with a stiffness of 120 kN/mm are used, leading to a 3 dB increase of the average pass-by sound pressure level.

We have performed dynamic mechanical analysis (DMA) tests on EVA and PU samples taken from a railpad and a typical vulcanized rubber used for railpad prototypes. The result is shown in Fig. 1. The storage modulus is increasing at low frequencies, but reaches a fairly constant value for both materials around 500 Hz for EVA and rubber. PU has a very low loss factor and shows no noticeable increase in storage modulus. The rubber's loss factor is clearly much higher, reaching values up to 1.5. The Poisson's ratio is difficult to determine. We assume a standard value of 0.3 for both EVA and PU, and a higher value of 0.485 for the nearly incompressible rubber.

3. DETERMINATION OF THE DYNAMIC STIFFNESS OF RAILPADS

3.1 Experimental setup

To ensure representative values for the dynamic stiffness, the pad is placed on a concrete sleeper and clamped under a steel mass. The foot of the mass is designed so that the standard Vossloh W14 fastening system can be used. The steel block has a mass of 10.6 kg, equal to that of a rail with 60E1 profile of the same length.

The steel mass is excited vertically by a shaker, using a swept sine signal ranging from 20 to 2000 Hz. The input force and vertical acceleration of the mass are measured using an impedance head. Assuming the steel mass behaves as a rigid body, and that its motion is limited to the vertical direction, the measured frequency response function gives direct insight into the pad's stiffness.

3.2 Finite element simulation

To gain more insight in the experimental parameters and the behavior of the pad, a digital twin of the experimen-

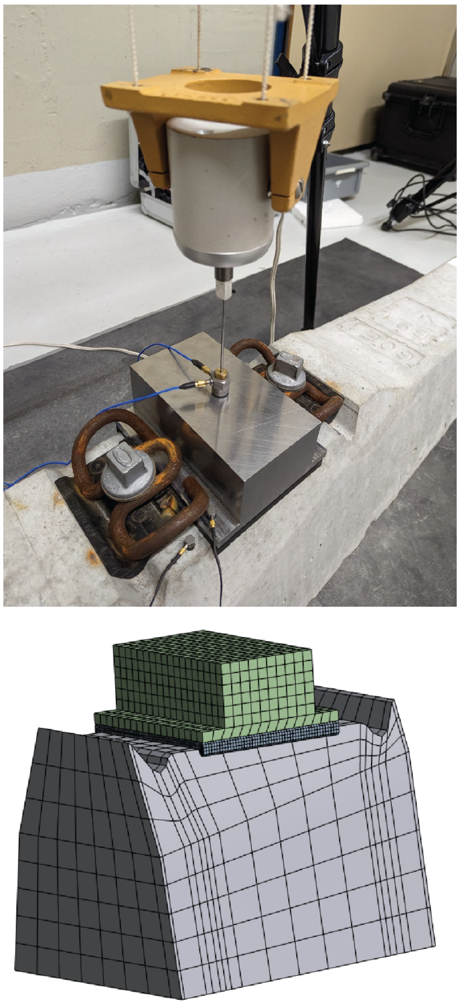

Figure 2. Top: experimental setup to measure the frequency-dependent stiffness of the railpads. Bottom: finite element model of the experimental setup, showing a detailed view of the fine mesh used for the pad.

tal setup was created using the commercial finite element modeling (FEM) software suite Ansys2022R2.

The frequency-dependent material properties can be introduced using an APDL script in order to match reality. To allow a linear harmonic analysis, the pad's surfaces are bonded to the supporting sleeper and to the steel mass. The resulting vertical displacement of the top mass as a result of a 1 N force excitation is calculated and used to extract the vertical spring stiffness.

3.3 Particle swarm optimization

Assuming the pad behaves as a massless spring and the sleeper and mass are rigid, the frequency response for the displacement of the mass-spring system can be written as

$$X = \frac{1}{k - m\omega^2} \,, \tag{1}$$

where the spring constant is complex $k = k_\mathrm{r} + \mathrm{i}k_\mathrm{i}$, m is the mass and $\omega = 2\pi f$ the angular frequency. Having a closer look at the results shown in Fig. 1, the rapidly increasing stiffness at low frequencies yields an initial decrease in displacement amplitude. At higher frequencies, a resonance peak is identified. This behavior justifies not using the standard measurement method for dynamic stiffness [6], which is a one-number value based on the identification of the resonance frequency alone. A constant stiffness would not explain the richer dynamics we identified.

In order to quantify the frequency-dependent spring stiffness, we propose to fit the unknown stiffness $k(f)$ by a smooth function with an increasing trend:

$$k(f) = k_{0,\mathrm{r}}(f-1)^{\alpha_\mathrm{r}} + \mathrm{i}k_{0,\mathrm{i}}(f-1)^{\alpha_\mathrm{i}} \,, \tag{2}$$

where $k_{0,r}$ and $k_{0,i}$ are the real and imaginary static values for the stiffness.

There are 4 parameters to fit the complex function. At this point, we limit ourselves to fitting the magnitude of the frequency response function (FRF) using particle swarm optimization. To do this, we use the standard function `particleswarm` implemented in Matlab R2020b, using 200 particles per iteration. This leads to a repeatable result which follows the simulated FRFs accurately. For the hard EVA pad, the results are $k_{0,\mathrm{r}} = 4.03 \times 10^8$ N/m/Hz, $\alpha_\mathrm{r} = 0.047$, $k_{0,\mathrm{i}} = 7.86 \times 10^7$ N/m/Hz, $\alpha_\mathrm{i} = 0.001$. The soft PU pad yields $k_{0,\mathrm{r}} = 4.99 \times 10^7$ N/m/Hz, $\alpha_\mathrm{r} = 0.001$, $k_{0,\mathrm{i}} = 7.43 \times 10^5$ N/m/Hz, $\alpha_\mathrm{i} = 0.30$. The frequency-dependent stiffness is shown in Fig. 4. The hard pads show a strong change with frequency, whereas PU pads show a rather constant stiffness.

4. CONCLUSION AND FUTURE WORK

We have developed a measurement setup to extract frequency-dependent stiffness values for railpads with strong visco-elastic behavior. The result shows a strong increase of the EVA pad stiffness, whereas its imaginary part remains constant. PU pads show a fairly constant

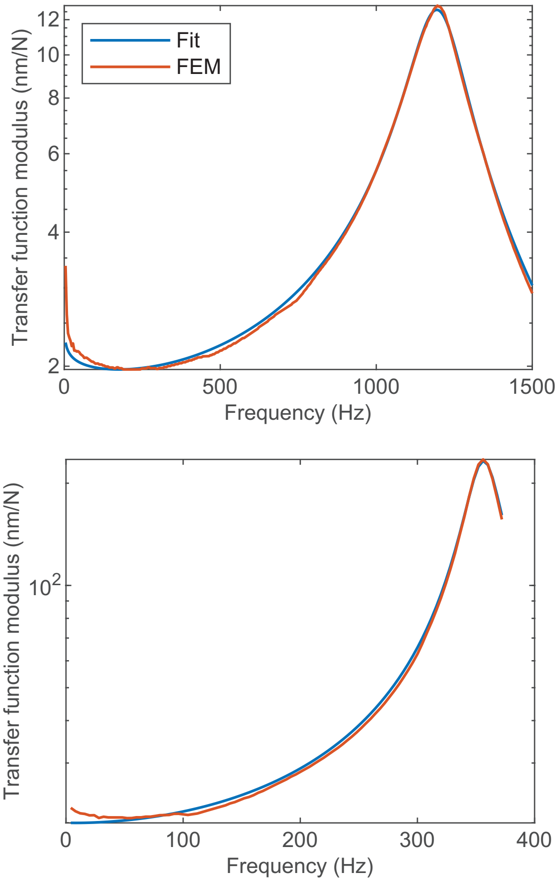

Figure 3. Simulated FRF (displacement/force) for an EVA (top) and PU (bottom) pad. Softer pads lead to higher displacement values. The curves can be fitted (blue line) by a frequency-dependent stiffness according to Eq. (2).

stiffness value over the entire frequency range of interest. Pads with a stiffness which increases with frequency allow to combine a reduced noise generation (stiff behavior at high frequencies) with lower vibration transmission (soft behavior at low frequencies). It is therefore beneficial to assess the dynamic stiffness over the entire frequency range.

In the next steps of this project, we will perform experiments on pads made out of different materials. This approach will eliminate the need for DMA measurements, and will directly yield the necessary input data for track models.

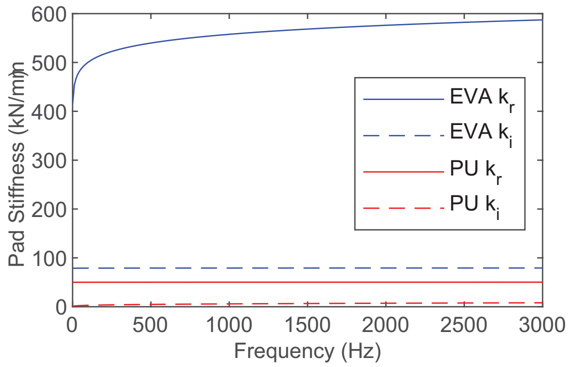

Figure 4. Real and imaginary parts of the pad stiffness for EVA and PU pads, calculated by fitting the FRF.

5. ACKNOWLEDGMENTS

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