DOI: 10.61782/fa.2023.0784

# ACOUSTICAL OPTIMIZATION OF RAILPADS BY F.E. MODELLING AND VALIDATION BY MEASUREMENTS

**E. Verhelst** [1*] **E. Balmes**[2]
[1] SD&M (Structural Dynamics and Monitoring), 1, Impasse La Borderie, 87120 Domps, France
[2] SDTools, 44, Rue Vergniaud, 75013 Paris, France

## ABSTRACT

Noise emission is often influenced by resonance. In case of wheel-rail contact during train pass-by, the rail is excited at his resonance frequencies, e.g., the pinned-pinned resonance. The only component in contact with the rail that can dampen this resonance is the rail pad. It is known that, when changing railpad dynamic properties as stiffness and damping, the pass-by emission can be reduced significantly. Several projects in Europe show that railpad optimization in terms of stiffness and damping results in noise reduction of more than 3 dB compared to the standard soft pad, for passenger and freight trains at speeds between 80 and 160kph. The goal of this study is to further optimize the rail pad design, to keep the flexibility in the track for protection of sleepers and ballast, combined with a high TDR (Track Decay Rate), or rail vibrational damping. The focus lies on the optimization of the stiffness distribution over the railpad surface. It is shown that, for the same global vertical railpad stiffness, the TDR can be optimized in the pinned-pinned area. The optimization is first done in a frequency dependent F.E.M. (Finite Element Modelling), then optimized railpads are mounted in a track and noise emission is compared with non-surface optimized railpads.

**Keywords:** *railpad, TDR, F.E.Modelling, optimization*

[*]***Corresponding author***: *eduard.verhelst@sd-m.fr.*


## 1. INTRODUCTION

Railway traffic will increase in the coming decades. This also means that railway traffic noise annoyance will become an issue, especially in urban areas. Noise mitigation measures are quite expensive and require dedicated planning and design in an early stage of new projects. For existing projects, applying noise reducing measures is much more complicated and very expensive.
Today we know that railpads can have an, not to be neglected, influence on the noise emission of railway traffic. There are projects where the railpad optimization in terms of stiffness, damping and design results in noise reduction of more than 3 dB for passenger trains and freight trains at speeds between 80 and 160 kph. [1].
Innovative railpad manufacturers, combining their knowledge of pad material and design with the knowledge we have today on Track Decay Rate or TDR [2], including the TDR-relation with noise emission, can easily produce new types of railpads that fulfil two requirements. One is keeping the flexibility in the track and protection of the substructures as sleepers and ballast, the other one is having a TDR or rail vibrational damping that is as high as possible. So, finding the ideal compromise between these two will lead to a future noise reduction of 3dB and more with low additional costs compared to other noise reduction measures.

## 2. METHODOLOGY

The methodology is based on the use of measurement data from a short length test track, combined with 2 Finite element models: a 2-D model, a classical ballasted half-track model and a more refined 3D-model. The optimization process is done in 3 steps:
Step 1: Defining and optimizing the needed stiffness and damping of the railpad to reach specific TDR requirements

and track engineering requirements, by combining test track data and the 2D – model. Also, temperature effects can be understood by measuring TDR at different temperatures. [3]
Step 2: Further optimize the geometry of the pad by using the 3D-model considering the acoustics requirements and track load requirements.
Step 3: Based on results of Step 1 and Step 2, the final component selection and mold geometrical dimensions are defined.

### 2.1 Defining an appropriate test track

It was decided to construct a short length ballast test track with UIC60 rails, Vossloh fixation, and concrete monoblock sleepers. (see Fig.1).

**Figure 1**. Short test section for TDR measurement.

That, easily accessible, test track could then also be used to measure the TDR as defined by the EN15461 on the designed railpad prototypes. Another big advantage of such an easily accessible test track is that temperature influence on TDR can be studied without having to go into a real track that then has to be put out of service. But first, the dynamic behavior of such a test track should be compared with a real track by means of finite element modelling.

### 2.2 Finite element beam model (classical ballasted half-track model)

Simulations, using a classical ballasted half-track model finite element model (see Fig.2) are done, to verify the effect of a limited length test track on the specific TDR results. Such a model is ideal to run batches for tuning the model with measurement data, since a dynamic response calculation takes only a few minutes.
But it is obvious that a 'limited' length track responses differently than a real 'infinite' length track. Depending on the damping and stiffness of the railpad, the TDR values for the 2 cases, especially in the higher frequency ranges above 1 kHz, are different. Tuning of the classical ballasted half-track model with measurements also learned that the introduction of a rotation stiffness between rail and sleeper was needed to get the proper pinned-pinned frequency. Probably not only the railpad, but also the rail fastener system, seems to have a significant influence on the dynamic behavior of the track.

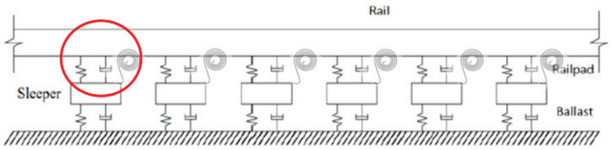


**Figure 2**. Finite element beam model used for verifying limited length test bench versus infinite track on TDR calculation.

A model is built, both for a short test section, as for a (nearly) infinite length track with 120 sleepers. 120 sleepers seem to be enough to cope with the side effects the accelerances (rail response for impulse excitation), needed for TDR calculation, can disturb. All element properties such as mass, springs, dampers, etc. are carefully defined. For the railpads, both the stiffness and the damping are defined as frequency dependent in a range from 10Hz to 5kHz. This is as close as possible to relevant data from test stands for measuring dynamic stiffness and damping where, in addition, an appropriate static preload was applied to the railpads. It has been found that a rotation spring between the sleeper and the rail was needed to tune the pinned-pinned mode of the rail, to avoid the railpad properties differ too much of realistic frequency dependent values. Thus, tuning the pinned-pinned frequency only to railpad properties seems to result in unrealistic values.
During the definition of the classical ballasted half-track model, in every possible step, the correlation between calculations and measurement is verified on all components. The calculated accelerances are compared with measured accelerances for the 2 cases for different rail pads. From these, after applying the EN15461, the TDR can be calculated. The indicated railpad stiffnesses are the static stiffnesses kSP in kN/mm. (according to EN13146-9:2020). The dynamic stiffness depend on the rail pad under study.

Figure 3 shows the calculated accelerances (mid-sleeper, direct), both for a short test section and an infinite length track (resp. red and blue curves). Also, the influence for a soft (100kN/mm) and stiffer (500kN/mm) pad are shown. The indicated resonances and anti- resonances are all verified by measurements for the 4 cases.

The next step is shown in Figure 4: the calculated TDR, by using the accelerances as defined in the EN 15461 for narrow band and 1/3 octave bands (Hz).

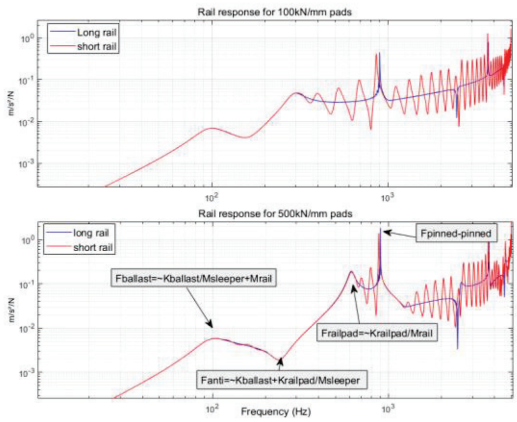


**Figure 3**. Accelerances calculated from the Finite element beam model for a short test section and an infinite length track for soft (upper plot) and stiffer (lower plot) railpads.

Discussion on the beam model accelerances and TDR results for 3 railpads: soft (100kN/mm), medium (300kN/mm) and stiffer (500kN/mm) railpads, with different damping values, (see Fig. 4):

- The accelerances function differs completely for both track lengths. This difference is, amongst others, influenced both by stiffness and damping properties of the railpad and also by the limited length.
- The pinned-pinned frequency is influenced by the type of rail, (mass/m, Inertia moments), by the railpad vertical stiffness, and by the rotational stiffness of the rail fixation. The latter is also influenced by the railpad stiffness and geometry.
- For soft pads, the shift in pinned-pinned frequency is more important.
- The effect of these shifts on the TDR is different depending on the damping of the pads: for highly damped pads, the testbench response is closer to an infinite track response, however for soft and less damped pads, there will be a serious overestimation of TDR above 1 kHz (e.g., 1 to 7 dB/m) and an underestimation between 500 and 1000Hz.
- This should be taken into account when designing the pads.

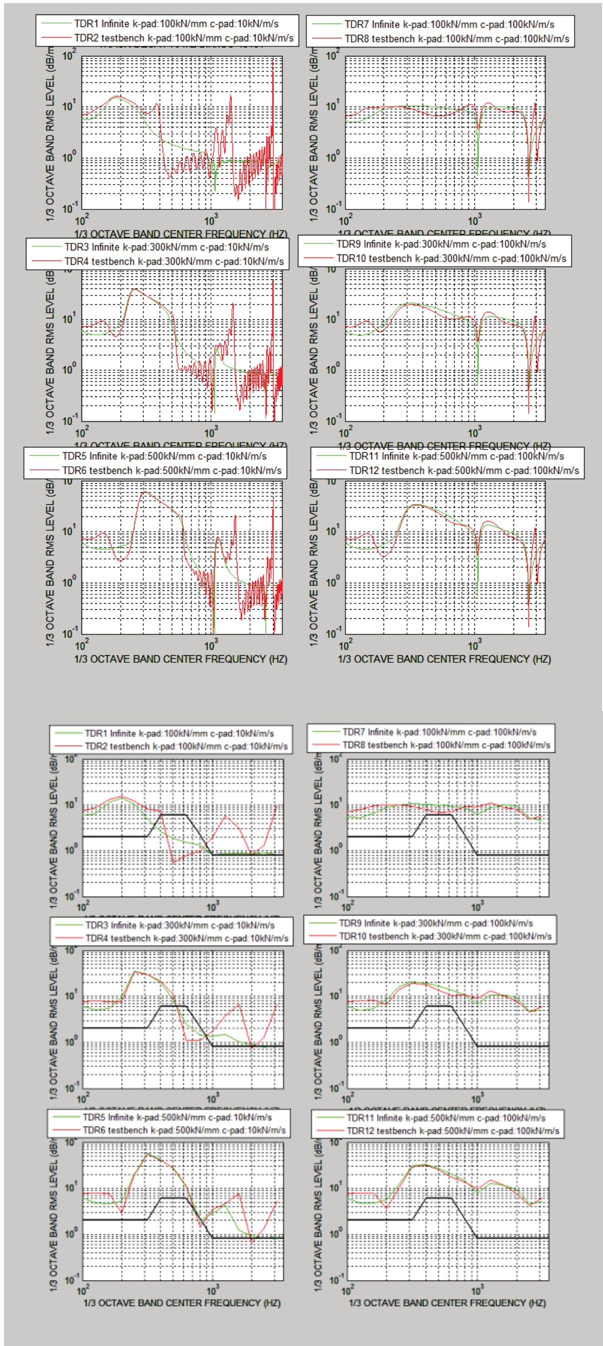


**Figure 4**. TDR calculated from the Finite element beam model for the short length (red curves) and an infinite length track (green curves) in narrow band and in 1/3 octave bands.

### 2.3 Finite element solids model

To analyze the vibroacoustic response of the track, while allowing for parametric changes of the pad properties, a 3D model, illustrated in figure 5, is used. A limited track section is modeled in SDT [4] using volume elements.

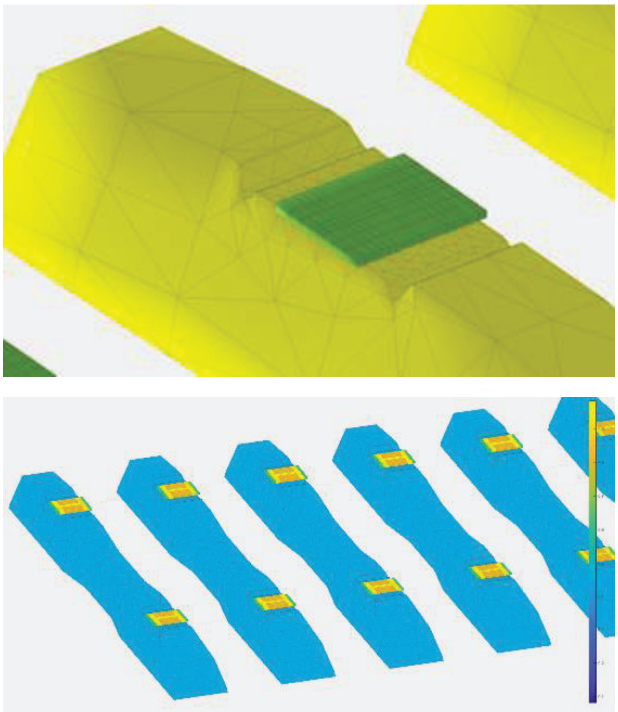

**Figure 5**. Solids model details showing the design of mesh to be able to distribute the stiffness over the railpad.

The following properties are used.

- A uniform stiffness density is distributed uniformly below the sleepers to achieve a ballast stiffness of 75 MN/m. A loss factor of 10% is associated with this stiffness as an approximation of energy radiation in the soil. More accurate representations require a soil model [5] which is beyond the objective of this study.
- Sleepers are modeled using as elastic volumes with E=30 GPa,$\nu$=0.25 and$\rho$=2500 kg/m$^3$, ignoring the steel reinforcements
- Rails follow the UIC60E1 profile and use E=210 GPa,$\nu$=0.3 and$\rho$=7850 kg/m$^3$
- Pads are modeled as viscoelastic materials with a frequency dependent complex modulus $E(\omega) = E'(\omega)(1 + i\eta(\omega))$. As typical pad tests characterize a frequency dependent stiffness rather than modulus, the property of linear dependence of stiffness matrices to constitutive properties is used thus the pad stiffness is a considered with a scalar coefficient$E(\omega)/E(0)$ describing the frequency dependence in amplitude and phase

$$K_{pad}(\omega) = \frac{E(\omega)}{E(0)}[K_{pad}(0)] \tag{1}$$

The dependence used is illustrated in figure 6. It is worth noting that more complex models could account for non-linear material effects. [6]

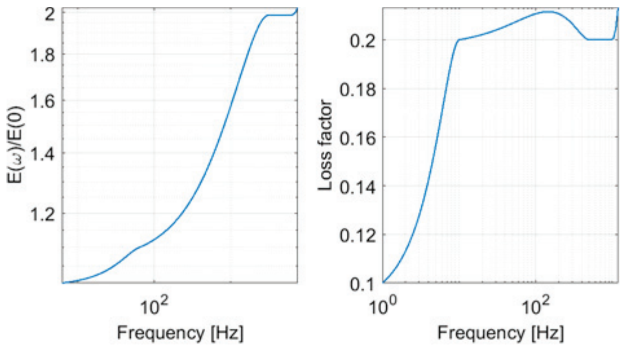


**Figure 6**. left : scaling coefficient$E'(\omega)/E(0)$, right : loss factor$\eta(\omega)$

- To allow optimization, the surface is divided into a grid of elements of which each element may have a different property. Top and bottom layers are also considered. For the design phase, the same frequency dependence is used for all elements, even though actual material tests for different constituents should be used in a verification phase.

Once the needed stiffness and damping properties for getting a specific TDR in combination with track design requirements are known, next step in the optimization process is to refine the model by using solid elements (see Fig. 5).

Then the effects of railpad geometry modifications, stiffness distribution over the railpad surface, etc. can be assessed before manufacturing the expensive molds to produce the pads. Through the whole process, a systematic validation of the model is done by measurements (accelerances by impulse excitations), also from parts separated in free-free suspension. Two models have been built. They include both a frequency dependent E-modulus and Loss factor for the rail pads. The short model has been built with 48.243 nodes and 77.780 elements and the 120-sleeper model with 417.588 nodes and 648.125 elements.

The meshing of the sleepers, rail and railpad is done in a way that the stiffness distribution over the railpad surface can be adapted to go to an optimal and high TDR for a defined global railpad stiffness. The distribution is first defined in a percentage per railpad solid element. In the next postprocessing step the global stiffness is calculated and adapted to result in a specific stiffness value, e.g. 100kN/mm at 1 kHz. Figure 8 shows the mode-shape of the short section model at the pinned-pinned frequency 1069 Hz.

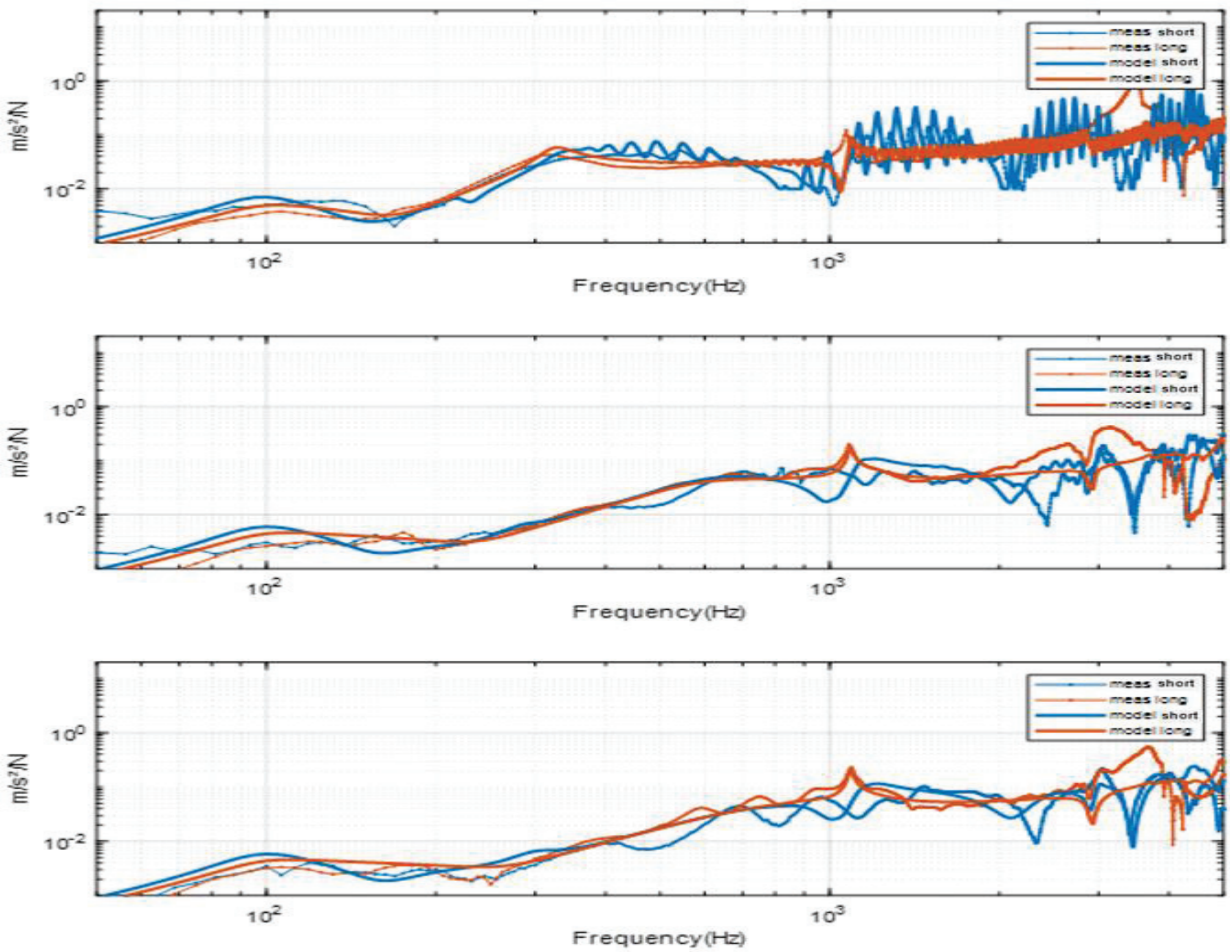


**Figure 7**. Beam model calculated accelerances (thick lines) comparison during tuning process with measurements, for 3 different railpads (with stiffness and damping variations).

## 3. RESULTS

### 3.1 Beam model (classical ballasted half-track model)

Figure 7 shows a comparison during the tuning process between the calculated and measured accelerances for 3 types of railpad, with different global stiffness, material composition and design.

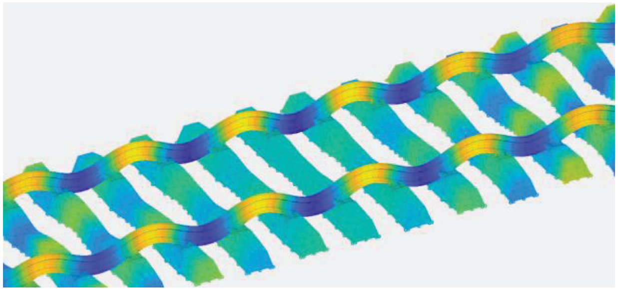

**Figure 8**. Solids model showing the mode shape at the pinned- pinned frequency.

During the tuning process, both the frequency dependent stiffness and the frequency dependent loss factor of the railpads are optimized.
We cannot expect to calculate the same narrow band accelerances as the measured ones for several reasons. In the classical ballasted half-track model, the vibration modes of the sleepers are not considered. The length and mounting of the rail, especially the ends surpassing the last and first sleeper also differ slightly. This can have an effect in the anti-modes, who are difficult to tune in the model. Also at frequencies above 2.5 kHz, effects of different accelerometer mounting can be a cause of differences.
But, the most important region to draw conclusions lies between 800 and 2kHz in 1/3 octave bands, and not in narrow band accelerances.

### 3.2 Solids model

The model as described in 2.3 is now used to optimize the stiffness distribution over the pad surface. An example of an optimization process given in figure 9. It can be seen that the TDR can be optimized even with small design changes in the stiffness distribution, especially in the region around the pinned-pinned frequency.

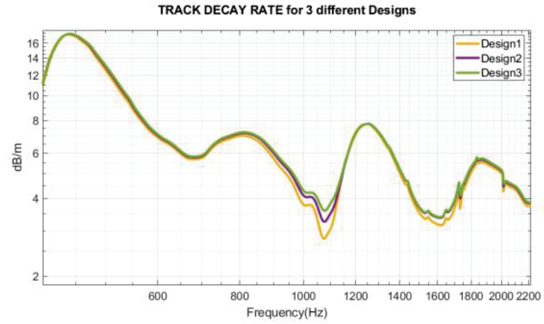


**Figure 9**. Sensitivity of the TDR on different railpad stiffness distribution, keeping the material composition and global vertical stiffness similar.

For this specific example, the final global stiffness and loss factor of the selected material was similar for the 3 cases, only the geometrical stiffness distribution over the pad is changed and optimized, taking into account also the technical specification as defined by track engineering. Several calculation batches finally result in the ideal stiffness distribution over the pad. The change in the TDR is maybe small but will lead at the end to more efficient rail pad, meaning higher TDR, and thus lower noise emission for a comparable amount of material and production cost.

### 3.3 Validation of the designed railpad

The designed optimized railpad was produced and installed in a test track and compared in terms of spectral pass-by noise (L(A)eq,tp) with a soft (red curve in figure 10) and a medium soft railpad (blue curve). The emission of geometrically optimized railpad is represented by the green curve. The kST stiffnesses of the medium soft and the geometrically optimized pads stay in the same range in terms of qualification, but the on-site measured TDR @1kHz differs nearly a factor 2. The TDR difference is not present at 800Hz and 1.25kHz. So, the optimization has a dominant effect purely on the pinned-pinned frequency. (also seen in the 3D-model). The optimization process results in a further noise reduction of the pass-by noise between 1 and 2 dB(A). The reductions are better than what could be expected from the TDR difference seen in the calculations.

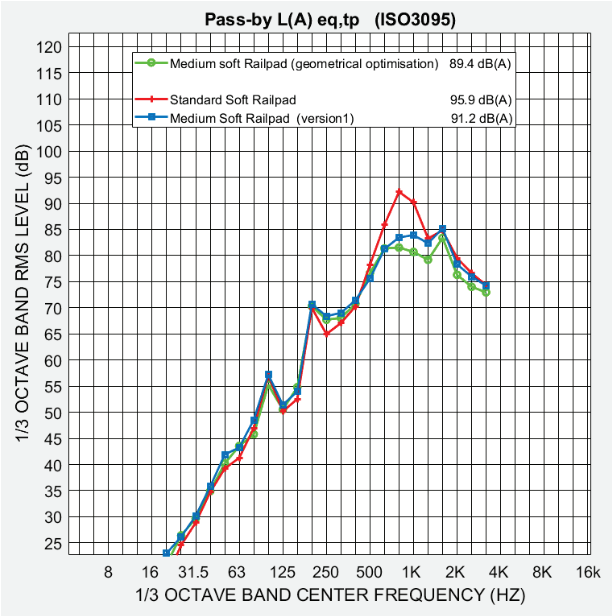


**Figure 10**. Comparison of 3 different railpads: soft (red curve) medium soft (blue curve) and the geometrically optimized railpad .

## 4. CONCLUSIONS

Optimizing a rail pad for a higher TDR and thus noise reduction should be more than just changing its stiffness. In the past, it was often mainly optimized in this direction. E.g., in reference [1] it was seen that for the Belgian case a reduction of 3 dB and more was feasible, of course, compared to a soft low-TDR railpad. The approach explained in this study goes two steps further.

By installing and using TDR measurements on a limited length test track in combination with 2D- and 3D finite element modelling, a three-step optimization of the railpad can be done much more efficiently.

The 3-D finite element model outcome (step2), proposing an ideal stiffness distribution over the railpad, combined with a modification of stiffness / loss factor (step1&3) relation leads today, to more acoustically efficient railpad designs. The final goal, a railpad, as soft as possible, with an as high as possible TDR can be developed to defined customer requirements. Even if the geometrical optimalisation effect on TDR is small, with all small steps summed up, there is a future potential to produce relatively soft railpads, that even have a better/higher TDR than extreme stiff but zero damping EVA railpads. In that way both track engineers as, acoustical engineers can be served. The track design requirements, resulting in good protection of the track substructure, can be finally combined with the very limited contribution of the track to the total rolling noise emission during train pass-by.

## 5. REFERENCES


[1] W. Verhelst, E. Heylen, “Development, validation and roll out of noise reducing rail pads on the Belgian INFRABEL railway network”, *IWRN13.*

[2] EN 15461+A1: Railway applications - Noise emission - Characterisation of the dynamic properties of track sections for pass by noise measurements.

[3] G. Squicciarini, D.J. Thompson, M.G.R. Toward and R.A. Cottrell, *The effect of temperature on railway rolling noise*, Institute of Sound and Vibration Research, University of Southampton

[4] *Structural Dynamics Toolbox (for use with MATLAB)* Paris: SDTools, 1995-2023. http://www.sdtools.com/help/sdt.pdf

[5] H. Pinault, *Réduction par apprentissage multi-nombres d’onde pour les guides d’ondes ouverts ou hétérogènes : application à la dynamique de la voie ferrée*, ENSAM thesis, 2020. https://pastel.archives-ouvertes.fr/tel-03131802

[6] R. Penas, E. Balmes, and A. Gaudin, “A unified non-linear system model view of hyperelasticity, viscoelasticity and hysteresis exhibited by rubber,” *MSSP*, vol. 170, p. 25, 2022, doi: https://doi.org/10.1016/j.ymssp.2021.108793.

[7] ISO 6721-1:2019: Plastics — Determination of dynamic mechanical properties — Part 1: General principles.

[8] E. Balmes, SDT/Visc, *Viscoelastic vibration toolbox, User Manual*. SDTools, 2004.