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# CALIBRATION OF TRANSFER FUNCTIONS ON A STANDSTILL VEHICLE FOR ON-BOARD INDIRECT MEASUREMENTS OF RAIL ACOUSTIC ROUGHNESS

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## ABSTRACT

On-board measurement of the acoustic performance of railway tracks is nowadays necessary to qualify networks on a large scale. The issue concerns not only the supply of prediction models for strategic noise mapping, but also the optimization of track maintenance. One of the key parameters to be measured is the rail acoustic roughness. Unlike direct measurements where sensors are directly applied to the rail surface, indirect measurements of rail roughness focus on quantities that result from wheel/rail interaction, such as noise or vibrations of axle-boxes or rail, and from which the effective wheel/rail combined roughness are estimated. In particular, on-board measurements make the qualification of long track lengths possible without major constraints on traffic. A number of improvements can be made to the existing methods, especially in estimating the transfer functions between the effective roughness and the signals provided by the sensors. This study is part of the MEEQUAI French project aiming to combine modelling and measurements to optimize the estimation of the transfer functions and the location of sensors while taking into account the variability of tracks. This paper concerns the measurement performed on a static vehicle/track configuration in order to validate and calibrate the numerical simulations.

## 1. INTRODUCTION

Rail is a mode of transport with a low environmental impact and acting for the modal shift from road to rail may be a powerful lever in the fight against global warming. However, this attempt may be hampered by some externalities, especially environmental railway noise. Rolling noise due to wheel/rail contact is the main source in a wide speed range. It is related to irregularities (roughness) of the wheel and rail surfaces that generate vibrations during contact [1]. Vibrating structures, in particular the wheel, rail and other track components radiate noise. The contribution of the track can thus be significant in terms of excitation and noise radiation. It depends mainly on the roughness of the rail, which is a function of its wear and grinding, and on the dynamic behaviour of the track, in particular the rates

of decrease of vibration energy in the rail (Track Decay Rate), which is primarily a function of the track supporting structure. The assessment of the acoustic performance of the track therefore requires the characterisation of these two parameters. This is true for the realization of the noise maps imposed by the European regulations, since the contributions of track and vehicle are now distinguished in the new common method CNOSSOS-EU [2]. It is also true in a perspective of acoustic maintenance of the track, for example to optimize rail grinding. Direct measures benefit from a normative framework in European regulations (EN 15610 and EN 15461) but they are tedious and costly as they require the intervention of qualified personnel and an interruption of traffic for several hours. Indirect on-board measurement is thus a promising alternative to qualify large-scale networks. Only roughness measurement is addressed in this paper.

As part of the MEEQUAI project, a comprehensive survey of existing on-board measurement systems was carried out [3]. Three families of methods were identified: (i) indirect measurements using accelerometers on axle boxes, (ii) indirect measurements from microphones in the wheel or bogie area and (iii) other types such as optical methods. Indirect methods based on vibro-acoustic sensors (i) and (ii) and transfer functions (see for instance [4, 5]) offer encouraging performance but suffer from a number of limitations. The first limitation concerns the frequency validity range which varies according to the type and the location of the sensors. The second limitation is related to the accuracy of the transfer functions, most of which are determined experimentally in a calibration step. This calibration is insufficient to account for variations in the transfer function when certain parameters deviate too far from the reference configuration, in particular parameters controlling the dynamic behaviour of the track and operating parameters.

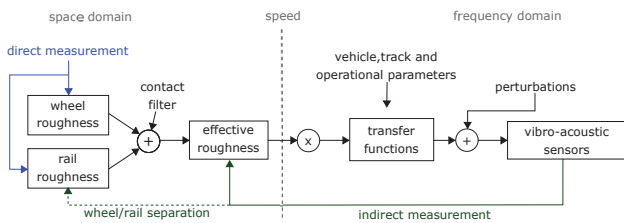
Two ways of improvement are proposed in the continuation of Chartrain's work [6]. To extend the frequency validity domain, a proposed solution is to combine different sensors whereas to increase the accuracy of the transfer functions, a method combining modelling and experimental calibration is developed, allowing to take into account

track and operational parameters [7]. The first experimental calibration of the transfer functions on a standstill vehicle is presented in this paper. The method for estimating the transfer functions and the different calibration options are first explained. The experimental campaign is then described. Finally, some results concerning the method based on axle-box accelerometers are presented and discussed.

## 2. A HYBRID METHOD FOR ESTIMATING TRANSFER FUNCTIONS

### 2.1 Overview of the indirect measurement method

Unlike direct measurements where sensors are directly applied to the rail surface, indirect measurements of rail roughness focus on quantities that result from wheel/rail interaction and from which the effective wheel/rail combined roughness is estimated (see Fig. 1). The rail roughness is then accessible, provided that the wheel and rail roughness can be separated and the contact filter can be properly evaluated. Indirect measurements are carried out in the frequency domain using on-board vibro-acoustic sensors (accelerometers and micropophones). Assuming a linear relationship between roughness and sensors, the effective combined roughness is obtained by an inverse process based on frequency transfer functions. A hybrid method based on modelling and experimental calibration is proposed for the precise estimation of the transfer functions [3, 7].



**Figure 1.** Schematics of the indirect measurement method

### 2.2 Wheel/rail interaction

The first step is the determination of the forces due to roughness at the wheel/rail contact points of an isolated wheelset, by using point dynamic receptances of both wheels  $[A_w]$ , rails  $[A_R]$  and contact  $[A_C]$  (see [1] for instance). The second step is the determination of the corresponding quantity on the sensors  $\{S\}$  by means of transfers between contact forces and sensors  $[H_{SC}]$ . Coupling between the vertical and lateral directions as well as between the two wheels/rails are considered, requiring the calculation of  $4 \times 4$  receptance matrices and leading to the following expression:

$$\begin{aligned} \{S\} &= [H_{SC}][A]^{-1}\{R\} \\ \text{with } [A] &= [A_w] + [A_R] + [A_C] \end{aligned} \quad (1)$$

where  $\{R\}$  is a vector specifying the roughness at both contact points (zero for lateral DoF's). The interaction model has been validated by comparison with the TWINS

software. Numerical models have been developed for the calculation of the wheel and rail receptance matrices as well as transfer matrices  $[H_{SC}]$  (see below). Contact receptance matrix is obtained using linearized Hertz models for normal contact and Thompson dynamic creep models for tangential contact [1].

### 2.3 Vibro-acoustic numerical models

Numerical Finite Element (FE) models have been developed in the 50 – 5000 Hz frequency range for several track configurations, and for a single vehicle type corresponding to a "CORAIL" test vehicle of SNCF-AEF (Railway Test Agency). This vehicle is used to conduct the first rolling tests in the framework of the MEEQUAI project. The set of tracks includes various types of layer (ballast/concrete slab), rail-pad dynamic properties (stiffness and damping), rail profile and sleeper configuration (concrete/wood, mono-/bi-bloc).

The FE structural models of the tracks are composed of solid elements for rail and sleepers, while rail-pads and ballast are modelled as discrete springs. In order to simulate an infinite rail length, "anechoic terminations" are included: rail material damping is gradually increased when moving towards the rail ends. The FE structural model of the complete wheelset includes brake discs, axle-boxes, bearings and control rods. All components are discretized using solid elements except for the bearings, which have been taken into account by means of equivalent stiffness. Track models have been validated by comparison with analytical models available in TWINS software [1] whereas the wheelset model has been updated by conducting an experimental modal analysis performed on a "free" wheelset resting on elastic supports.

Exterior sound radiation models of wheelset and tracks use Finite Elements for the free-field combined with infinite elements or Perfectly Match Layers for the far-field. They include reflections on the platform and in the bogie area through impedance boundary conditions. A sufficient track length is taken into account in order to verify the theoretical effects of the longitudinal directivity of the field.

### 2.4 Calibration strategy

Despite all the attention paid to model development, indirect measurement requires calibration of the transfer functions. An important advantage of the proposed hybrid method is that the calibration may only cover the transfer functions  $[H_{SC}]$  which are mainly dependent on the vehicle and not on the track or operating conditions. Two calibration options are considered. Calibration during rolling on a reference track is classical but requires direct measurement of roughness. Another approach consisting in performing the calibration on a standstill vehicle is presented in the following sections.



### 3. EXPERIMENTAL CAMPAIGN

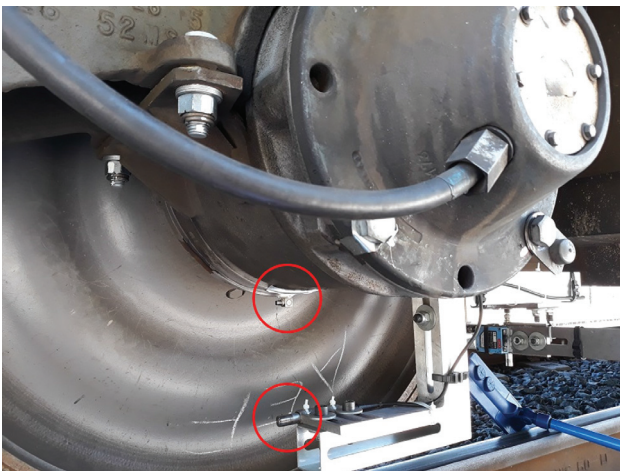
#### 3.1 Tested on-board sensors

The selection of the sensors that could be tested in the project was made on the basis of the developed numerical models. The evaluation of the possibilities of indirect measurement offered by several families of sensors, particularly in terms of frequency range, and the quantification of the effects of the dynamic characteristics of the track and of the operating parameters on the transfers made it possible to optimise the number and location of candidate sensors [3, 7]. Table 1 lists the different selected sensors located on both sides of the wheelset. The target frequency band is also indicated.

Sensor class	Number	Range (Hz)
Axle-box accelerometers	2	50 – 2000
Near-field microphones		
close to wheels	6	1000 – 5000
close to rails	2	500 – 2000
Under-coach microphones	4	500 – 5000
Under-coach MEMS	11	500 – 5000

**Table 1.** Tested on-board sensors

The test vehicle was instrumented with the sensors listed in Table 1. Specific supports have been designed by SNCF-AEF to allow the measurement during rolling at speeds up to 160 km/h in a second phase (see Figs. 2 to 5).



**Figure 2.** Axle-box accelerometer and axial microphone close to the wheel

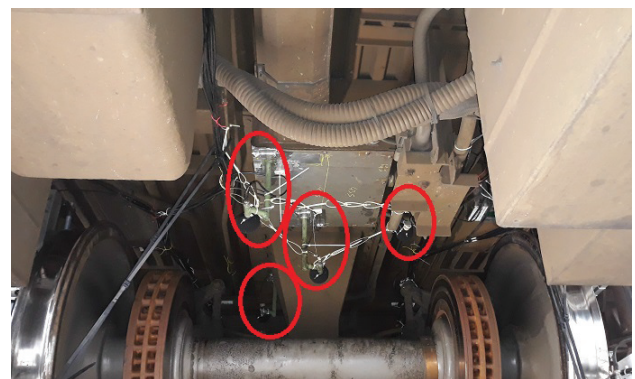
#### 3.2 Calibration procedure

##### 3.2.1 Track dynamic behaviour and transfers to microphones close to the rail

The measurements were carried out on a ballasted track representative of the tracks encountered on the French national network, consisting of UIC60 rails resting on bi-bloc concrete sleepers and 9 mm thick rail-pads (*a priori* medium stiffness). The first phase consisted in measuring the vibro-acoustic response of the track (alone) in the



**Figure 3.** Supporting structure for near-field microphones (wheel and rail)



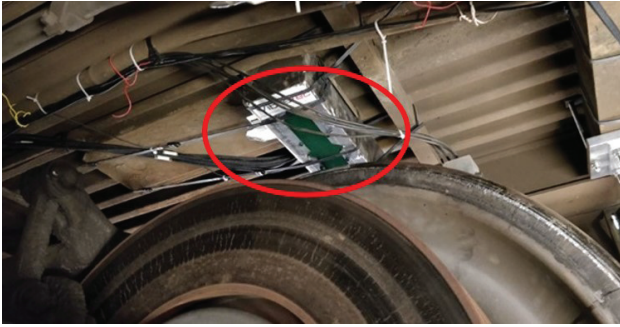
**Figure 4.** Under-coach microphones

range 50 – 5000 Hz to vertical and lateral point excitations using an impact hammer. For the acoustic response, microphones were positioned close to the rail, up to 6 m from the excitation point (see Fig. 6). This was useful in order to calibrate the vibratory model of the track but also the rail radiation impedances (ratio between the pressure and the rail vibration in front of the microphone). In particular, it appeared that the rail-pads were much stiffer than expected, i.e. around 1000 MN/m instead of 350 MN/m, the value generally observed for this type of pad. The reason for this difference seems to be related to the strong tightening of the fasteners.

##### 3.2.2 Transfers to axle-box accelerometers

The second phase consisted in calibrating the transfers to on-board sensors, in the presence of the instrumented vehicle standing on the track, using point excitations with the impact hammer. According to the families of sensors, different procedures were used.

For axle-box accelerometers, given the great uncertainty concerning the influence of the applied load and the mounting of the wheelset and its dynamic behaviour (especially on bearing stiffness), the objective was to directly calibrate the transfer function  $[H_{SC}]$  in the 50 – 2000 Hz range on a mounted and loaded wheelset for a better accu-



**Figure 5.** Under-coach MEMS assembly



**Figure 6.** Microphones close to the rail for the calibration of rail acoustic impedances

rary. However, two issues occur for this calibration.

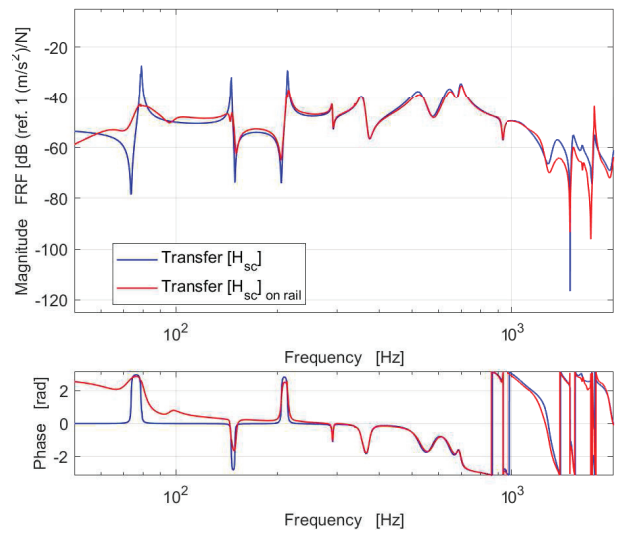
First, the measurement on a loaded/mounted wheelset is only possible when the vehicle is resting on the track, thus including the effect of coupling with the track. This effect was quantified by using an impedance matching method and the models presented above. It can notably be shown that:

$$[H_{sc}]_{\text{on rail}} = [H_{sc}][A]^{-1}([A_R] + [A_c]) \quad (2)$$

The results in Fig. 7 show that the effect of track coupling is not fundamental but significant. A damping at resonances of the global axle modes is observed, especially below 400 Hz, as well as in the vicinity of the radial wheel mode with 2 nodal diameters, i.e. around 1500 Hz. The procedure adopted therefore consists in measuring the transfer functions in the presence of the track  $[H_{sc}]_{\text{on rail}}$  and correcting them by inverting Eq. (2):

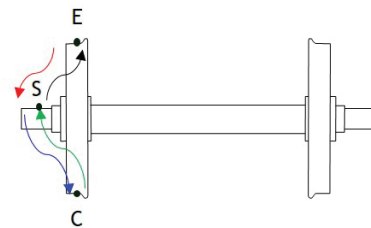
$$[H_{sc}] = [H_{sc}]_{\text{on rail}}([A_R] + [A_c])^{-1}[A] \quad (3)$$

The second issue is the application of the vertical excitation, as the wheel tread is not accessible due to the presence of the track (transfer to be measured in green in Fig. 8). Two alternatives have been considered: (i) vertical excitation on the wheel tread, at  $180^\circ$  from the wheel/rail contact point, under the assumption of symmetrical behaviour of the wheel in the frequency range of interest (transfer in red in Fig. 8) and (ii) vertical excitation on the axle-box and vibrations recorded on the wheel at the wheel/rail contact



**Figure 7.** Effect of the coupling with the track on the transfer from vertical wheel/rail contact forces to vertical axle-box vibration (numerical results)

point using two tri-axial accelerometers, under the assumption of reciprocity (transfer in blue in Fig. 8).



**Figure 8.** Alternative measurement of the transfer from wheel/rail contact force to axle-box vibration

### 3.2.3 Transfers to microphones close to the wheel, under-coach microphones and MEMS

Calibration of transfers to microphones close to the wheels was performed only in terms of wheel radiation impedances (ratio between pressure and vibration response of the wheels at the microphones) and at the resonances of the main wheel modes involved in rolling noise in the range 1000 – 5000 Hz (radial modes and axial modes with 1 nodal circle). Different excitations were used (vertical and lateral, at wheel/rail contact if possible or elsewhere on the wheel tread) depending on the modes targeted and the quality of the measured vibro-acoustic transfer functions (coherence) was monitored.

A similar method has been adopted for the calibration of transfers to under-coach microphones and MEMS. However, the radiation impedances were calculated relative to the vibrations measured at the wheel/rail contact point.



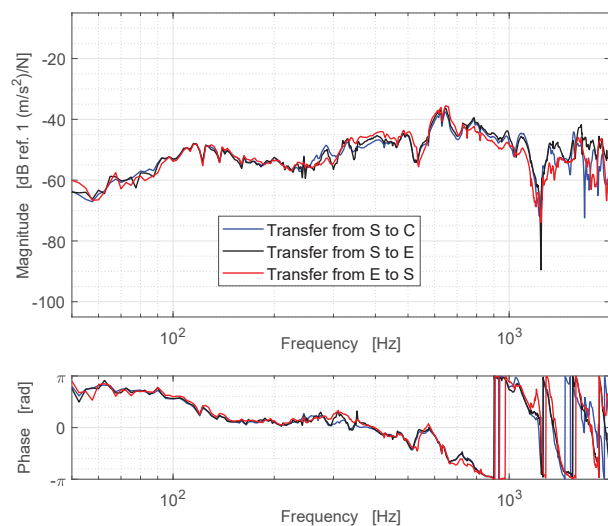
## 4. RESULTS FOR AXLE-BOX ACCELEROMETERS

In this section the results concerning the transfers to axle-box accelerometers are presented and discussed. The coherence of the transfers is satisfactory for most of the measurements, except below 300 Hz for the cross-transfers between the two sides of the axle. In this range, the low vibration levels are probably filtered by the axle-box bearings. In a concern of synthesis, only the transfers in the vertical direction are presented, which is the most important for drawing conclusions.

### 4.1 Excitation point and reciprocity

Fig. 9 shows the comparison between the responses at the contact point (point C, blue transfer in Fig. 8) and at the point at 180° from the contact (point E, black transfer in Fig. 8), to an excitation on the axle-box (point S). The response at point E is very close, in magnitude and phase, to the one at contact, up to about 1200 Hz. In this frequency range, it is thus possible to excite the wheelset at point E, if the contact point is not accessible, and this without an alteration of the transfer that one wishes to measure.

The validity of a reciprocal measurement of the transfer to the axle-box is also examined on Fig. 9 by comparing the direct (red in Fig. 8) and reciprocal transfers (black in Fig. 8). Reciprocal transfer is very close, in magnitude and phase, to direct transfer up to about 1200 Hz. In this range, reciprocal action may be thus taken if direct action is not possible, and this without loss of information on the transfer that one wishes to measure.



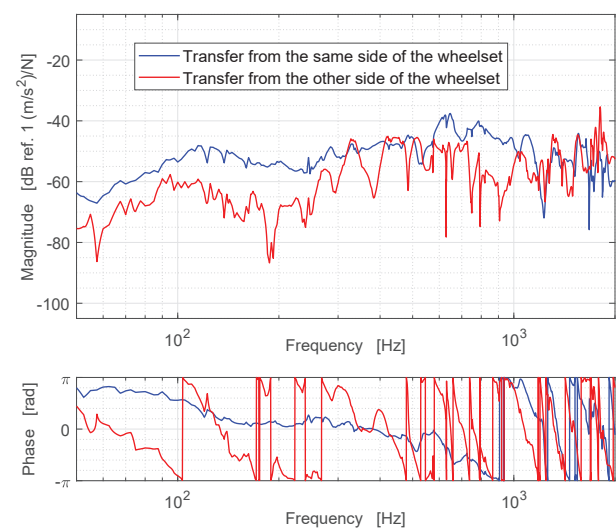
**Figure 9.** Comparison of alternative measurement of vertical transfers from wheel/rail contact force to axle-box vibrations (track effect included)

### 4.2 Coupling between the two wheels

The estimation of the coupling level between the transfers from the forces on the two rail lines is essential for the inverse procedure in the indirect method. Indeed, a low

coupling level could justify a separate identification of the roughness on each rail, an assumption frequently made in existing on-board measurement methods.

Fig. 10 shows the comparison between the response on each axle-box to an excitation at one wheel/rail contact point (estimated by using reciprocity). Up to 1000 Hz, the response on the wheel on the opposite side of the excitation is about 10 dB lower than on the response on the same side, except in the range 300 – 600 Hz where the gaps are reduced and response may have similar magnitudes at some frequencies. Above 1000 Hz, the coupling is more accentuated. This coupling level seems to make it possible to identify the roughness distinctly on each rail line by neglecting the crossing terms in the transfer matrices of Eq. (1) only below 300 Hz and between 600 and 1000 Hz. For all other frequencies, the coupling has to be taken into account in the identification.



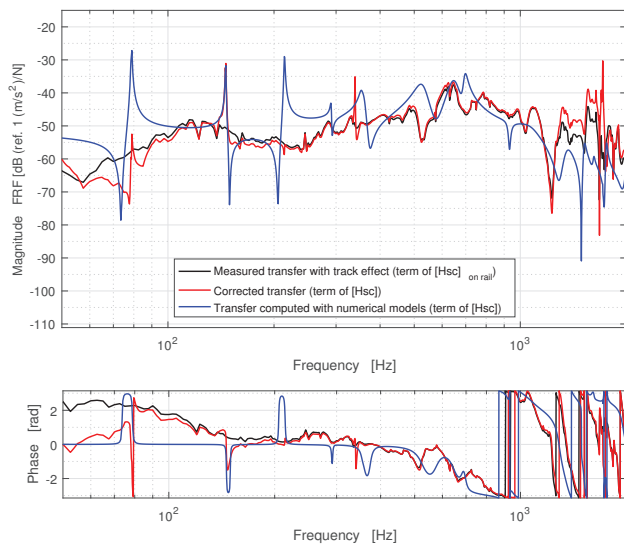
**Figure 10.** Comparison of measured vertical transfers from wheel/rail contact forces to axle-box vibrations on the same and opposite sides (track effect included)

### 4.3 Correction for track effect and comparison with the model

Using the alternative methods outlined in paragraph 4.1, the various transfers corresponding to the terms of matrix  $[H_{sc}]$  on rail (vertical/lateral on both sides of the wheelset and all crossing terms) were measured in the presence of the instrumented vehicle resting on the rail. These transfers were then corrected using Eq. (3) to obtain matrix  $[H_{sc}]$ .

For the vertical direction, Fig. (11) shows the effect of the correction and the comparison of the obtained experimental transfer with the transfer computed with numerical models. As expected, the correction makes certain resonances appear more clearly at low frequencies and significantly modifies the transfers above 1200 Hz. The comparison with the model is correct in tendency up to about 1200 Hz but there are some important differences such as the higher damping or the shift of some resonances. Above

1200 Hz the differences are more pronounced. These results show on the one hand the importance of the axle load and mounting effects, the model having been recalibrated only on the wheelset in free conditions, and on the other hand the interest of the proposed calibration procedure.



**Figure 11.** Comparison of measured and computed vertical transfers from wheel/rail contact force to axle-box vibrations

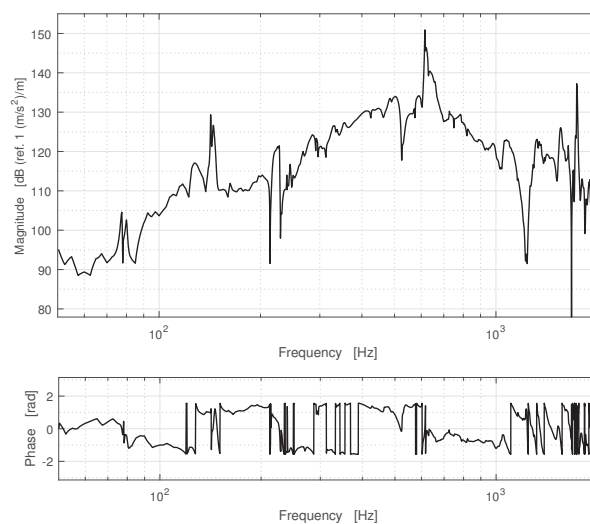
#### 4.4 Final roughness/sensor transfer

Once the contact/sensor transfer  $[H_{sc}]$  has been corrected for the track effect, it can be used to calculate the total transfer between roughness and sensor outputs (see Eq. 1). Rail receptances  $[A_R]$  varies with the excitation position, due to the periodicity of track numerical models. In rolling conditions, this position changes with time, and this should be taken into account for the hybrid calibration. As a first approximation, the effect of the position of the excitation has been taken into account through a quadratic average of global transfer functions relative to 2 positions: above a sleeper, and midway between two sleepers.

Fig. 12 shows the vertical acceleration on one axle-box, for a unit value of roughness on 1 rail line (same side of the axle-box). It should be observed that this transfer is not comparable to any measurement in rolling conditions, since there will always be contributions of roughness coming from both rail lines. For comparison and validation with rolling measurements, averaged quantities (both accelerations and roughness) for the 2 rail sides should be considered.

### 5. CONCLUSION

The article presents a brand-new methodology for on-board indirect roughness measurements, which is based on a multi-sensor and hybrid calibration approach. While numerical models were used to estimate the wheel/rail interaction and to optimize the sensor positions, experimental



**Figure 12.** Vertical transfer from combined roughness to axle-box vibrations

calibration is still needed for transfer functions that only depend on the vehicle characteristics.

An experimental campaign on a standstill vehicle has been presented, where a large set of sensors have been tested. During the campaign, transfers from the contact point to sensors were measured, thus allowing to draw a certain number of conclusions. It was observed that the coupling between the 2 axle-boxes of a wheelset is low, except in limited frequency ranges. Different alternatives for exciting the wheelset were analysed. It was concluded that reciprocal excitation at the axle-box, as well as excitation at a point  $180^\circ$  away from the contact, are acceptable alternatives to direct excitation. In the end, a correction for the track effect on the contact/sensor transfer has been presented and validated. It allows to obtain the transfer function to be injected in the interaction model, in order to obtain the final roughness/sensor transfer.

A final validation of the hybrid calibration will come from comparison with future rolling measurements.

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