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A METHOD FOR TRACK-INDEPENDENT RAILWAY VEHICLE CHARACTERISATION AS GROUND-BORNE VIBRATION SOURCES BASED ON BLOCKED FORCES

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ABSTRACT

In this article, a method to characterise rolling stock as sources of ground-borne noise and vibration is presented. The method is based on the blocked force approach, and it allows for a fully track-independent characterization of railway vehicles. It is designed for laboratory use, serving as a tool that rolling stock manufacturers can employ to assess the ground-borne vibration emissions of new railway vehicle prototypes. This can be achieved by conducting experiments at their own facilities using only a small track section and a set of excitation devices. This paper presents a numerical study of the feasibility of the proposed approach in which the particularities of the method are discussed through various simple case studies. Preliminary instructions to conduct experimental tests to characterise blocked forces of real rolling stock are also included.

Keywords: *Railway-induced vibration, Rolling stock, Source characterisation, Blocked force*

1. INTRODUCTION

In urban environments, implementing railway transportation systems in the underground space avoids airborne noise pollution, contributing to a quieter and more sustainable city while preserving surface space for other urban developments. However, underground railway traffic does disturb residents of nearby areas because of the so-called ground-borne vibration and the consequent structure-borne noise (commonly referred to ground-borne or re-radiated noise), which can be significant sources of annoyance in densely populated cities with large underground railway networks.

Research in railway-induced ground-borne noise and vibration has been intense over the past two decades. However, there is still a deficit of well-established standard procedures to predict, test and control the problem in comparison to airborne noise topic, for which standardisation is widespread and consolidated. This can be observed clearly in the case of experimental characterisation of rolling stock as source of noise and vibration. On the one hand, noise emissions from new and retrofitted rolling stock have been regulated under the European technical specification for interoperability for airborne noise emissions (Noise TSI) [1] since 2022. Because noise emissions are influenced by both track and rolling stock dynamic behaviour, a standardized test procedure has been established to minimise the influence of the track. To do so, Noise TSI sets limit curves for track decay rates and rail roughness of the track in where the vehicle certification is performed, in order to ensure that the differences on noise measurements taken at the reference microphone are mainly induced by different rolling stock designs and parameters.

On the other hand and in contrast to airborne noise, there is still no standardised test procedures to quantify the ground-borne vibration emissions of rolling stock using track-independent indicators. Nonetheless, some research has been recently conducted in this topic. In a recent work, Ntotsios et al. [2] proposed what they called

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as track-independent vehicle indicators (TVIs) to classify railway vehicles in terms of their ground-borne noise and vibration emission. The proposed indicators, one for ground-borne vibration and another one for ground-borne noise, are based on the force density at the railhead, which can be indirectly obtained from field measurements of railway-induced ground vibration using the line source transfer mobility at the test site. Both force density and line source transfer mobility are quantities described and used in the in the detailed vibration assessment guidance of the Federal Railroad Administration (FRA) and Federal Transit Administration (FTA) standards [3, 4]. The force density is chosen as the basis of the TVIs because, in contrast to the vibration levels at the ground surface, it is found to be relatively track-independent. Nevertheless, authors suggest that some restrictions should be applied to the site to avoid unwanted influence from the track or ground condition. The paper also proposes using transposition procedures to convert the results to a standard reference scenario aiming to reduce this influence. Both TVIs proposed in [2] are defined as the sum over all relevant frequency bands of the frequency-weighted force densities, being the frequency weightings defined to represent the human sensitivity to ground-borne vibration and re-radiated noise. Thus, different trains can be compared in terms of their vibration emission performance comparing the relative differences of their TVIs.

In another work, Villot et al. [5] preliminary proposed the use of blocked forces as a fully track-independent indicator to quantify the vibration emissions of rolling stock. The blocked force approach allows for characterising a vibration source fully independently to the receiver structure to which is attached. In [5], the vehicle-track system (including ballast) is considered to be the source, while the soil is assumed to be the receiver. Authors propose to determine the source mobility using the blocked forces and the free velocity of the source at the source/receiver interface, suggesting that blocked forces and free velocity quantities can be determined imposing extremely rigid and soft soils, respectively, as receiver systems, in the context on of a numerical assessment. The paper also suggests that FTA standard may be seen as an equivalent form of the indirect method for *in situ* blocked force characterisation [6]. Building on this preliminary proposal, the present work aims to further explore the applicability of blocked forces to quantify the vibration emission of trains in the basis of the indirect method of blocked force characterisation [6]. Here, to deploy the blocked force approach, the railway vehicle is considered to be the source, while the track-embankment-(tunnel)-soil system is assumed to be the receiver. To test this configuration, this paper presents a numerical study to illustrate and assess the blocked force approach. Initially, a model constructed in the framework of the moving roughness approach [7] is considered. This approach allows for estimating the ground vibration induced due to railway traffic considering the vehicle to be fixed (non-moving along the track), while the roughness is pulled through inducing an excitation to both the wheels and the rail at the contact points. This model provides a good approximation of the ground surface vibration levels when the train circulation speed is low [8]. The assumption that the vehicle is not moving allows for an direct application of the blocked force approach, as shown in the paper. However, the paper also explores the situation where the vehicle is assumed to be moving at a constant speed over the track, in which the calculation of the blocked forces is conducted in a moving frame of reference that follows the train motion.

2. MODELLING STRATEGY

The vehicle-track model schematically represented in Fig. 1 is adopted with the aim of numerically studying the applicability of the blocked force approach for characterising trains as vibration sources. The model consists of a

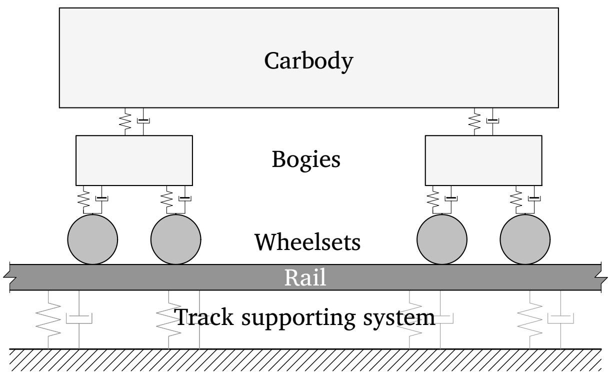

Figure 1. Global model layout.

vehicle with four axles resting over a simplified model of the track, which is based on an Euler-Bernoulli beam, as a model of the rails of the track, resting over a continuous supporting system consisting of longitudinally distributed spring and dashpot elements, representing in a simplified way the fastening systems and the embankment that supports the rails. This assumption is known to be approximately valid for the frequency range of interest in railway-induced ground-borne noise and vibration problems, from 1 Hz to 250 Hz [9]. It is assumed that both rails are equal, that are resting over the same supporting system and have the same roughness profile. Based on this, the system to be studied can be reduced to just one rail and half vehicle.

The vehicle model considers the carbody and the bogies to be rigid bodies experiencing vertical and rotational (pitch) motions, while the wheelsets are supposed to be particles that can only move vertically. Bogie and wheelsets are connected through spring-dashpot systems that represent the primary suspensions of the vehicle while spring-dashpot systems connecting the carbody and the bogies are representing the secondary suspensions. The contact between the wheels and the rail is modelled using the Hertz contact theory through an equivalent spring accounting for the linearised contact stiffness. Parameters and degrees of freedom (DOFs) of the vehicle system are detailed in Fig. 2.

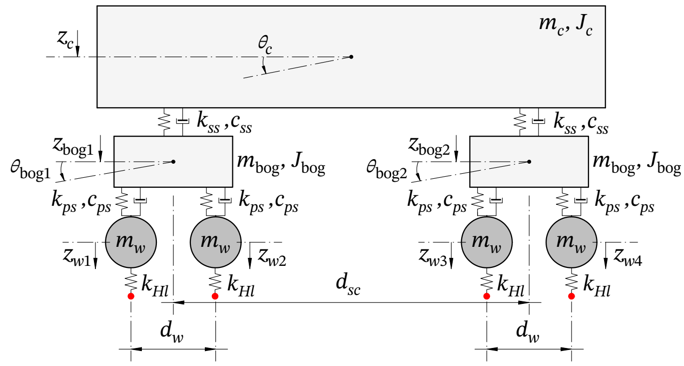

Figure 2. Vehicle model. Detail of the adopted DOFs and parameters considered. Contact points with the track, at the bottom of the contact springs, are denoted with red dots.

Thus, the equations of motion of the isolated vehicle system can be derived using rigid multibody dynamics, leading to the following expression

$$\mathbf{D}_v \hat{\mathbf{u}}_v = \hat{\mathbf{f}}_v, \tag{1}$$

where $\mathbf{D}_v$ represents the dynamic stiffness matrix of the vehicle model and $\hat{\mathbf{u}}_v$ is the vector that collects the displacements associated with the system's DOFs, being

$$\begin{aligned}\hat{\mathbf{u}}_v = \{&\hat{z}_c \;\; \hat{\theta}_c \;\; \hat{z}_{\text{bog1}} \;\; \hat{\theta}_{\text{bog1}} \;\; \hat{z}_{\text{bog2}} \;\; \hat{\theta}_{\text{bog2}} \\ &\hat{z}_{w1} \;\; \hat{z}_{w2} \;\; \hat{z}_{w3} \;\; \hat{z}_{w4} \;\; \hat{z}^v_{c1} \;\; \hat{z}^v_{c2} \;\; \hat{z}^v_{c3} \;\; \hat{z}^v_{c4}\}^{\text{T}},\end{aligned} \tag{2}$$

where $\hat{z}^v_{ci}$, for $i = 1, 2, 3, 4$, are the vertical displacements of the contact points in the vehicle model. Furthermore, $\hat{\mathbf{f}}_v$ is the vector that collects the forces applied to the vehicle system. The hat notation is used to denote variables in the frequency domain.

Regarding the track, the dynamic stiffness matrix $\mathbf{D}_r$ is constructed, accounting for the four DOFs associated to the wheel-rail contacts plus and extra one representing the vertical displacement at an evaluation point located at d_e from the left contact point. To this aim, the analytical solution of the response of an Euler-Bernoulli beam subject to harmonic loading is employed (see for example [10]). As an alternative to the modelling strategy based on the moving roughness approach, the train may be considered to be moving along the track through the moving train approach [11]. In this case, the roughness profile of the rails is attached to the track while the train is moving at a constant speed over it. To handle this problem, a moving frame of reference is introduced following the vehicle motion, in which the contact forces are determined. For the model adopted in the present work, the response of an infinite Euler-Bernoulli beam over a Winkler foundation subject to a moving harmonic load is employed to determine the receptance of the track in this moving frame of reference.

Vehicle and track sub-models can be combined to construct the coupled vehicle-track model by assembling the two dynamic stiffness matrices, reaching to the following system of equations

$$\mathbf{D}_C \hat{\mathbf{u}}_C = \hat{\mathbf{f}}_C, \tag{3}$$

where the subindex C refers to the coupled vehicle track model and where the total amount of DOFs is 15, the last one being the extra evaluation point over the track, which will be used for validation purposes.

As shown in Fig. 3, this paper proposes to apply the blocked force approach considering the source system to be the vehicle model presented in Fig. 2 and the receiver system to be the track-embankment-(tunnel)-soil system, in this case just represented by a flexural beam on a Winkler foundation.

In this context, the indirect approach to determine the blocked force can be formulated as [13]

$$\hat{\mathbf{u}}_{Cc} = \mathbf{H}_{Ccc} \hat{\mathbf{f}}_{Sc}, \tag{4}$$

where $\hat{\mathbf{u}}_{Cc}$ is the response of the coupled system due to the excitation induced by the roughness at the wheel-rail contact points, while $\hat{\mathbf{f}}_{Sc}$ are the blocked forces and $\mathbf{H}_{Ccc}$ is

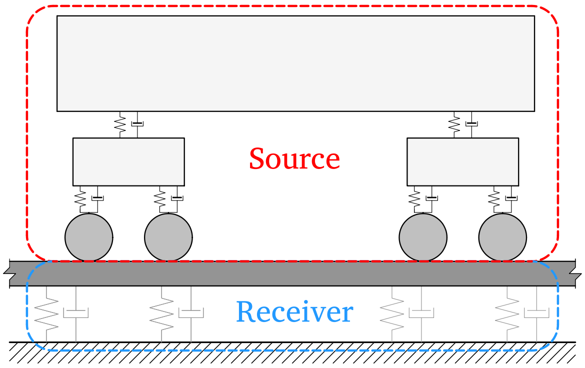

Figure 3. Source and receiver definitions considered in the framework of the blocked force approach.

the receptance matrix of the coupled system at the wheel-rail contact points. This expression is employed to determine the blocked forces in this paper. Then, the response on an arbitrary set of DOFs in the receiver can be obtained from the blocked forces as

$$\hat{\mathbf{u}}_{Ce} = \mathbf{H}_{Cec}\hat{\mathbf{f}}_{Sc}, \tag{5}$$

where $\hat{\mathbf{u}}_{Ce}$ is the response at the arbitrary set of points in the receiver and $\mathbf{H}_{Cec}$ is the receptance matrix that relates the response at those points with forces applied at the contact DOFs in the coupled system. This expression is also used in the present paper to ensure the correctness of the source description based on the blocked force.

To account for the excitation due to the roughness, consider a roughness profile that induces a deformation on the contact spring. This can be represented in the coupled model by external forces applied on the wheelsets and the rails as $\mathbf{F}_{wr} = k_{Hl}\mathbf{z}_r$, being $\mathbf{z}_r$ a vector that collects the roughness frequency spectra a the four contact points. Particularly, the forces applied to the coupled system are

$$\hat{\mathbf{f}}_C = \{0 \;\; 0 \;\; 0 \;\; 0 \;\; 0 \;\; 0 \quad -\mathbf{F}_{wr} \;\; \mathbf{F}_{wr} \;\; 0\}^{\mathrm{T}}. \tag{6}$$

Considering the correlation of the roughness between the wheels [8] and assuming a roughness profile that result in unitary forces, forces $\mathbf{F}_{wr}$ can be written as

$$\begin{aligned}\mathbf{F}_{wr} = \{&\exp(\mathrm{i}\omega x_{a1}/v) \;\; \exp(\mathrm{i}\omega x_{a2}/v) \\ &\exp(\mathrm{i}\omega x_{a3}/v) \;\; \exp(\mathrm{i}\omega x_{a4}/v)\},\end{aligned} \tag{7}$$

where x_{ai}, for $i = 1, 2, 3, 4$, are the positions, from the left wheel, of the wheel/rail contacts, ω is the angular frequency and v is the train speed.

Before conducting a numerical s

3. NUMERICAL VERIFICATION OF THE PROPOSED APPROACH

In this section, the results of the numerical study conducted are presented. Firstly, values adopted are listed in this paragraph. The numerical values adopted for the half vehicle are following. The car body has a mass $m_c = 22.8$ tones and a pitch moment of inertia $J_c = 55.6$ kg cm^2. The secondary suspension has a stiffness of $k_{ss} = 295$ N/mm and a viscous damping coefficient $c_{ss} = 21$ N s/mm. Each bogie has a mass $m_{\mathrm{bog}} = 2028$ kg and a pitch moment of inertia $J_{\mathrm{bog}} = 645$ kg m^2. The primary suspension has a stiffness of $k_{ps} = 700$ N/mm and a viscous damping coefficient $c_{ps} = 4.35$ N s/mm. Each wheelset has a mass $m_w = 482.5$ kg. The linearised Hertz contact stiffness k_{Hl} is set to be 200 kN/mm. The geometrical layout is defined by the distances between bogie centres ($d_{sc} = 15.2$ m) and between wheelsets on the same bogie ($d_w = 2.2$ m). The values adopted for the track parameters are also listed here. The rail cross-sectional area is $S_r = 7.690 \cdot 10^{-3}$ m^2, and its second moment of area is $I_r = 30.55 \cdot 10^{-6}$ m^4. The material properties of the rail are given by a Young's modulus $E_r = 210$ GPa, a density $\rho_r = 7850$ kg/m^3, and a structural damping factor $\mu_r = 0.01$. The supporting system is characterized by a stiffness per unit length k_f that takes two different values in this work: 500 (kN/mm)/m (referred as stiff support) and 25 (kN/mm)/m (referred as soft support). Corresponding viscous damping coefficients c_f of the supporting system are 3 (kN s/mm)/m and 0.3 (kN s/mm)/m, respectively. Finally, the distance d_e is set to be 5 m and the train speed v takes the value of 25 m/s.

To check the validity of the proposed method, the response at the evaluation point on the railhead is obtained using the previously defined roughness excitation as well as through the blocked forces for the case of the track with stiff support. The results are shown in Fig. 4, where a perfect agreement can be observed between the response obtained applying the excitation to the coupled system, via Eqn. (3), and the response obtained with the blocked forces, using Eqn. (4) to determine the blocked forced from the response at the contact points and then employing Eqn. (5). Results also show the response at the left wheel/rail contact point for procedure verification purposes. Once it is verified that obtained forces are properly representing the source, another verification is proposed to ensure they are actually blocked forces. Thus, a comparison of the forces obtained for two different receivers has been conducted, considering a track over stiff

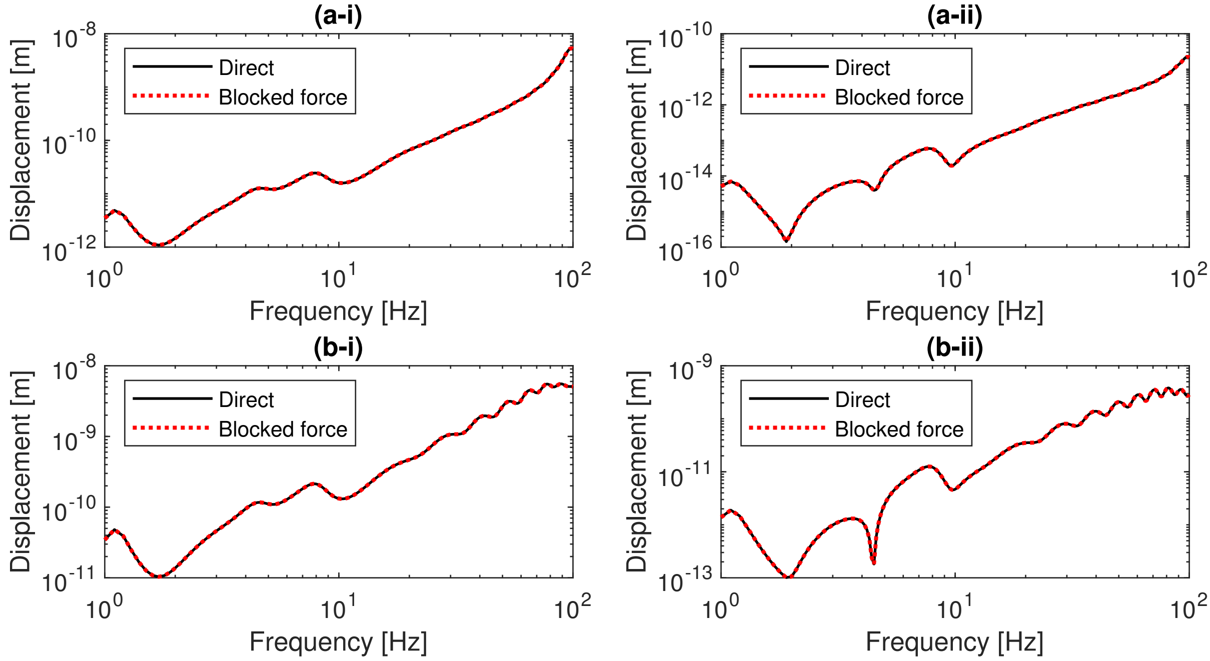

Figure 4. Response at left wheel/rail contact (a) and at the evaluation point (b) for the track with Stiff support (i) and soft support (ii) obtained using the direct calculation and through the blocked forces.

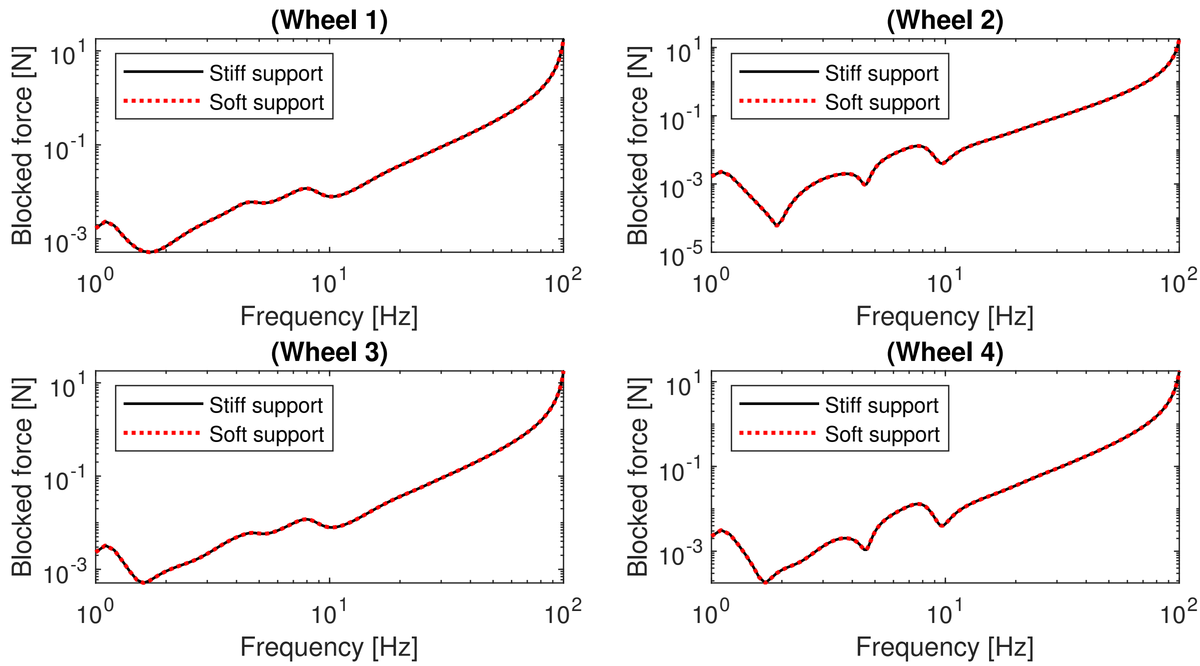

Figure 5. Comparison of blocked forces obtained considering the track with stiff support and soft support.

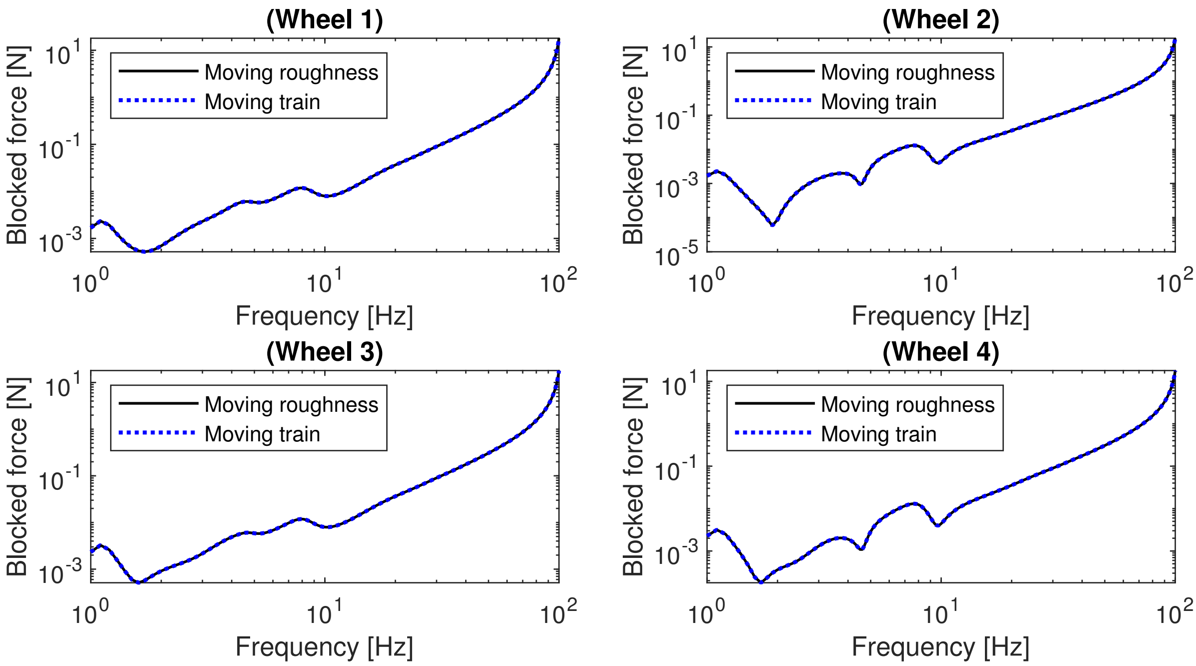

Figure 6. Comparison of blocked forces obtained considering the moving roughness and the moving train approaches.

support and a track with soft support. As expected, since all DOFs at the contact source/receiver interface are considered, blocked forces are equal irrespective of the track system to which their are attached, as shown in Fig. 5.

Finally, the blocked forces are determined in the context of the moving train approach, in a frame of reference following the train. As again expected, the resulting forces are equal to the ones obtained in the framework of the moving roughness approach, as shown in Fig. 6. This can be explained from the fact that the only change applied when determining the blocked forces in the moving train approach with respect to roughness one is the track receptance, and blocked forces are independent of the receiver considered.

4. PRACTICAL CONSIDERATIONS

This blocked force approach would be used to characterise new railway vehicle prototypes or retrofitted ones in terms of their vibration emissions at the facilities of railway track manufacturers. This can be achieved taking use of a small track section, in where the vehicle prototype should be placed, and a set of excitation devices that can be applied on the wheels and railhead to reproduce the excitation induced by the rail roughness. While this is not a fully realistic evaluation of the blocked forces, due to the simulation of the rail roughness excitation using, for example, electrodynamic shakers, it would allow for testing different track conditions directly in the laboratory, bearing in mind the random nature of the roughness in real tracks. As shown in the paper, blocked forces obtained in this context can be used to simulate the response in any track even considering the vehicle is moving. Problems related to the incompleteness of the interface DOFs may arise when deploying this experimental test.

5. CONCLUSIONS

In this paper, the potentiality of blocked forces as a track-independent indicator of the vibration emission of a railway vehicle is studied numerically. Using a theoretical model of the vehicle/track system, it is shown that blocked forces can uniquely define the vibration emission of railway vehicles irrespective of the track supporting system. Furthermore, blocked forces characterising a moving train can also be determined and they are exactly equal to the ones of a non-moving train. The results suggest that the approach is applicable to experimentally characterise railway vehicle in the manufacturers facilities and even *in situ*, although more research should be conducted in this regard to reach a robust testing methodology.

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