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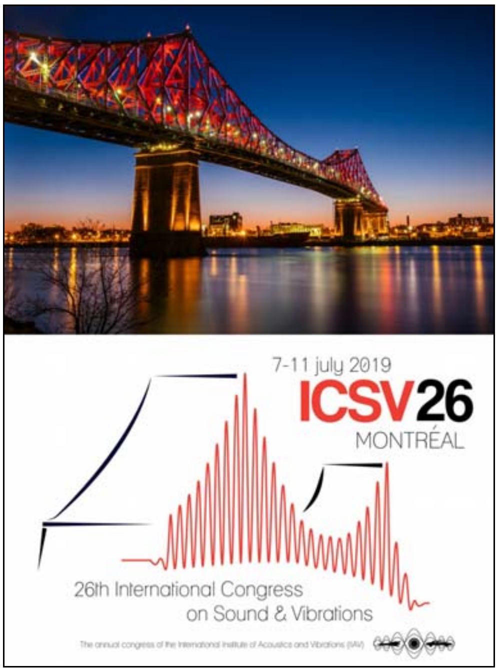

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RAILWAY GROUND VIBRATIONS AND DESIGNING DYNAMIC VIBRATION ABSORBERS FOR URBAN RAIL TRANSIT

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Dynamic vibration absorbers (DVAs) are widely accepted in vibration engineering as one of the most suitable ways to suppress undesirable low-damped resonance. In railway design, it is often placed on the track as a mitigation measure coupled to a floating slab. This paper presents the study of a tram vehicle equipped with DVAs with the aim to suppress the specific mode that affects the vehicle/track interaction and the surrounding ground wave generation. A complete vehicle/track/soil numerical model is developed and used to evaluate the dynamics of the whole system. Modal decomposition is applied to the vehicle/track model to obtain the optimal DVA parameter values for any location. Different DVA masses are analysed and different DVA locations are proposed. From the results, the vehicle bogie is revealed to be the best location and a compromise can be found for the DVA mass for reducing the ground vibration level of around 20%.

Keywords: dynamic vibration absorber, turnout, rail joint, ground vibration, Brussels tram

1. Introduction

The generation of railway ground-borne vibrations is a consequence of vehicles passing over tracks, generating dynamic forces from the rotating wheels into the track. These forces depend on the moving vehicle's load (close to the static contribution, often called quasi-static effect) and on the surface irregularities at the wheel and rail surfaces (representing the dynamic interaction between the vehicle and the track/soil subdomains). They both contribute to the propagation of vibrations outwards from the track [1]. Three steps are usually retained in the analysis of railway-induced ground vibration:

- **Predicting and measuring the ground vibration levels around specific railway networks.** Several studies were performed these last years, including the development of dedicated prediction tools [2–6] or intensive measurement campaign [7, 8]. Such studies allow collecting vibration data to be analysed and to determine if they exceed allowable thresholds.

- **Understanding the causes of elevated ground vibration levels.** This part is relevant and complex since many subsystems interacts each others. In urban area, the origin mainly arises locasized defects like joints, turnouts, switches [9, 10]
- **Proposing mitigation solutions to alleviate excessive vibration levels.** Actions were often proposed for the track [11, 12].

In recent years, Dynamic Vibration Absorbers (DVAs) have been widely used in railway tracks with the aim of attenuating low- and middle-frequency vibration and noise. Grassie and Elkins [13] showed that promising reduction in rail corrugation is obtained if a DVA is used to detune and damp the relevant mode of vibration. Thompson et al. [14] showed significant reductions of track component noise with a designed mass-spring–damper DVA with multiple tuning frequencies. Zhu et al. [15] compensated the amplification area associated to floating slab tracks using multiple DVAs judiciously tuned.

In [16], a first study was proposed to demonstrate that a DVA placed on a vehicle could prevent feelable vibrations in neighbour track. It was based on the use frequency response functions (FRFs) of a vehicle/track system. The studied case was the T2000 tram circulating in Brussels (Figure 1) for which abnormal vibrations were observed when passing on localized defects model and at low speeds. It was also observed that the tram bogie's bounce mode significantly contributes to the generated ground vibrations. The aim of this paper is to complete the results presented in [16] by analysis additional results and the possible reduction in the ground vibration levels. To do this, a complete vehicle/track/soil analysis will be performed, including the DVA dynamics in order to evaluate the potential of such mitigation measures in the case of urban conditions prior to any physical test.

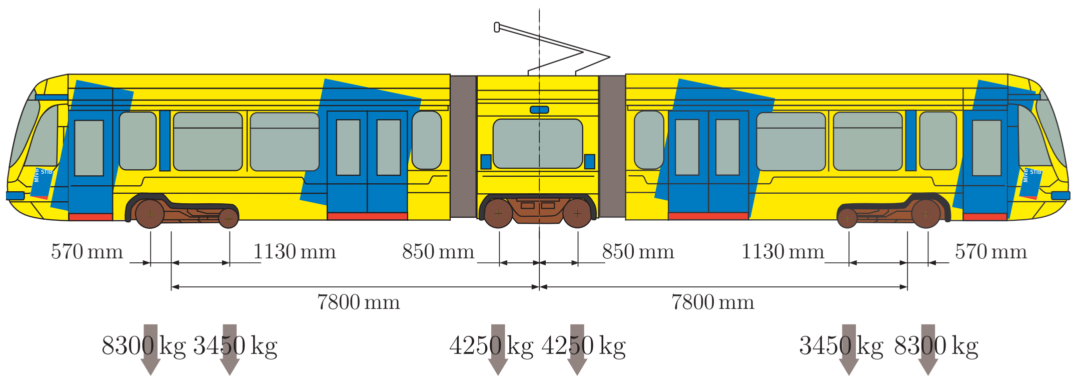


Figure 1: Main dimensions and axle loads of the T2000 tram.

2. Basic concepts of the proposed prediction model

The proposed prediction model is based on two successive calculations [9]. Complexity required the problem to be split. This approach allows the most well-suited modelling approach to be used for each subsystem. First of all, the dynamics of the vehicle/track subsystem is simulated by considering a multibody vehicle model (Fig. 2) moving at speed v_0 on a flexible track (Figure 3) with a rail irregularity. The wheel/rail forces are defined using non-linear Hertz's theory and allows the coupling between the vehicle model and the track. The latter is defined as a flexible beam (Young modulus E_r, geometrical moment of inertia I_r, section A_r and density ρ_r) discretely supported by the sleepers (of mass m and with spacing L), including viscoelastic elements for the ballast (stiffness k_b and damping coefficient d_b) and the railpads (stiffness k_p and damping coefficient d_p). To take into account the dynamic behaviour of the foundation, which plays an important role at low frequencies, a coupled lumped mass (CLM) model is added to the track model, with interconnection elements for the foundation-to-foundation coupling. The ballast reactions are then saved and used as input forces for the second step, which is devoted to the calculation of ground wave propagation using a finite element analysis.

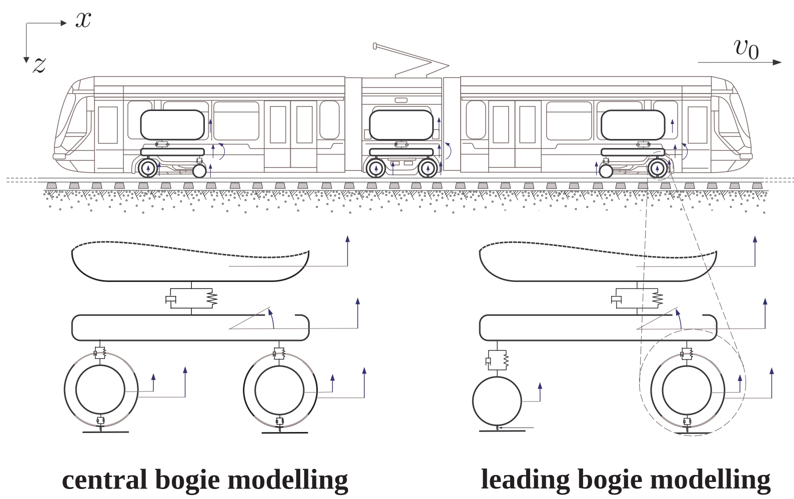


Figure 2: T2000 tram multibody model (blue arrows represent the different degrees of freedom).

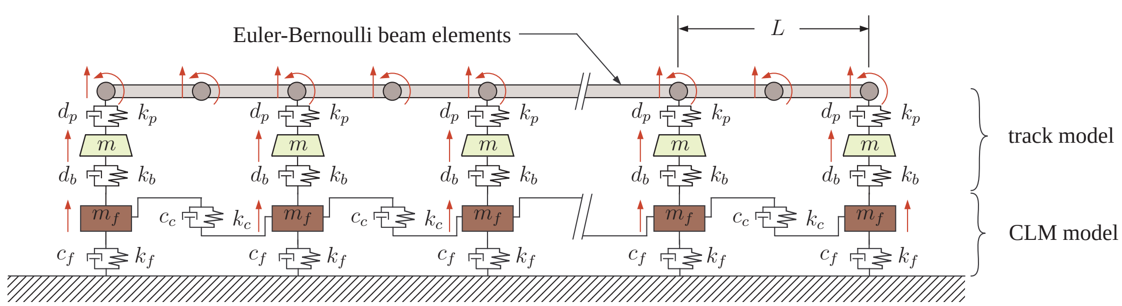


Figure 3: Track/foundation coupling.

The design procedure of a DVA for a vehicle/track system is based on the modal decomposition associated to the simple procedure for a single-degree-of-freedom system [17]. The DVA is identified by its mass m_a, spring k_a and dashpot d_a, associated to the motion in the same direction as the undesired motion of the primary system (Figure 4). Mathematically speaking, a one degree-of-freedom system is added either to the vehicle (case 1: DVA on vehicle) or the track (case 2: DVA on track). Both configurations can be analysed separately in order to confirm which solution is the most efficient.

3. Frequency response function analysis: optimizing DVA parameters

As the vehicle/track interaction primarily affects the ground vibration, it is proposed to focus on the FRF of motorized wheels (in front of the leading car bogie; a similar observation can be drawn for the central car bogie) with two different scenarios (a DVA placed on the bogie and a DVA placed on the track). For the latter, two configurations will be retained: a DVA placed on the rail or a DVA placed on a sleeper close to the localized defect.

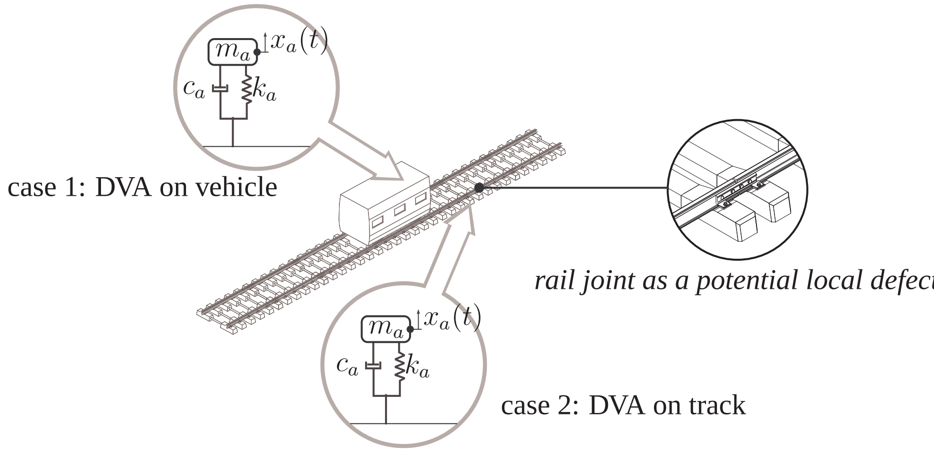


Figure 4: Schematic representation of a damped dynamic vibration absorber attached to the vehicle/track system.

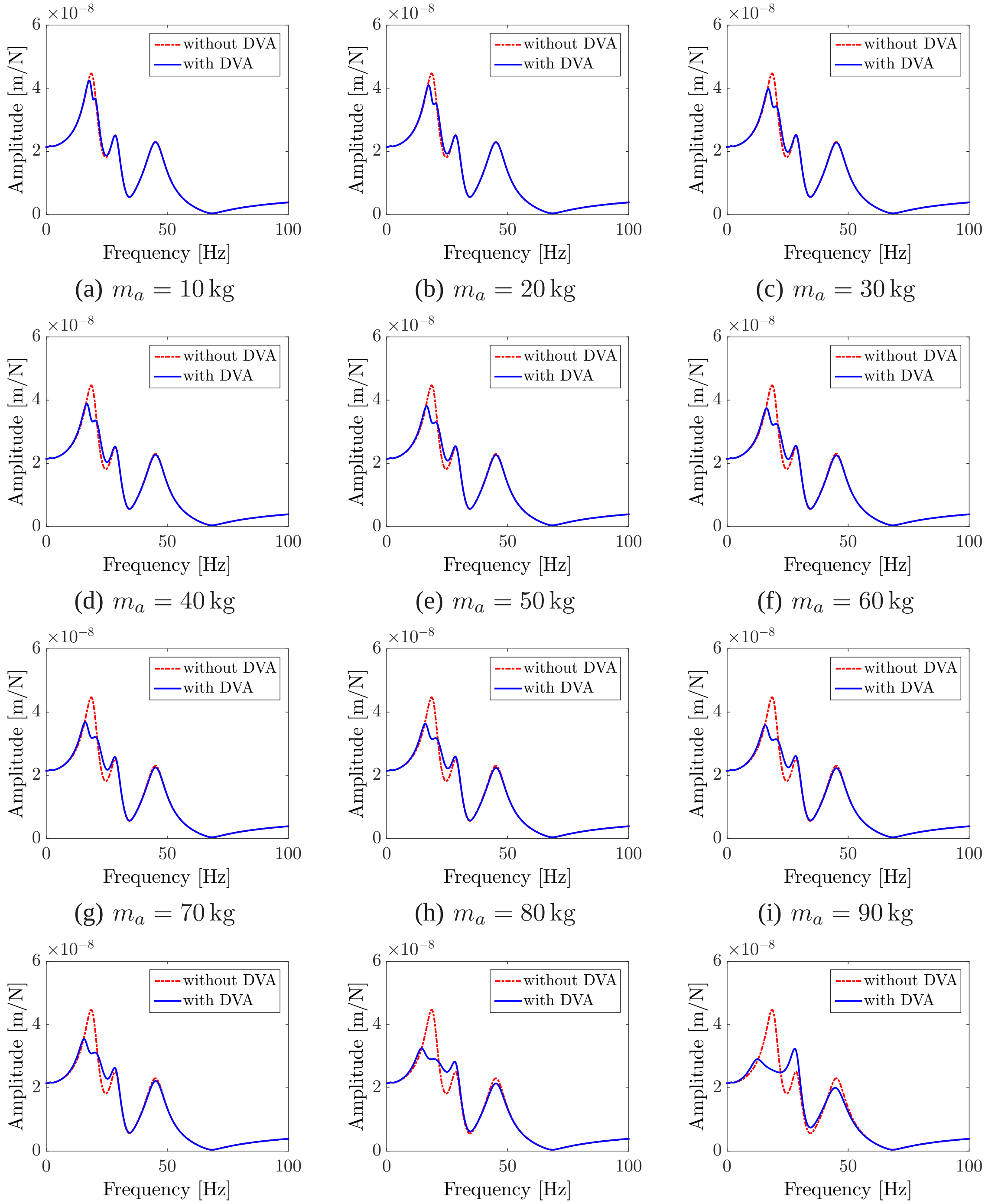
First scenario results are presented in Figure 5 which shows the calculated FRFs by varying the DVA mass m_a into a realistic range (stiffness k_a and damping ceofficients d_a are tuned to their optimal values for each case). It is observed that a reduction of peak amplitude at $19.6\,\mathrm{Hz}$ (corresponding to the eigenfrequency of bogie bounce mode) is gradually observed; however, for greater DVA masses, a negative impact to the next vehicle mode is observed, with a increasing of the peak amplitude of the bogie pitch mode (around $29\,\mathrm{Hz}$), cancelling the gain brought by the DVA on the first peak. An optimum value of $200\,\mathrm{kg}$ of DVA mass m_a is therefore deduced from this scenario.

Figure 6 presents a similar study regarding the second scenario. The studied range of DVA mass m_a is larger than previously, knowing that the track can support an important additional mass. It is shown that the gain is smaller: the target resonance peak is less reduced for both configurations (a DVA placed on the rail and a DVA placed on the sleeper). An amplification of the next resonance peaks (around $29\,\mathrm{Hz}$ and $45\,\mathrm{Hz}$) is also visible, but with maxima less pronounced. Finally, it is observed that for a DVA mass m_a of $2000\,\mathrm{kg}$, the peak amplitude exceeds the original value without DVA.

The conclusion for this study is that a DVA placed on the vehicle is more efficient than a DVA placed on the track. However, such analysis is based on the frequency domain and cannot take into account the realistic forces during the passing of the tram in the localized defect.

4. Ground vibration time history: validating DVA location

Figure 7 presents the results at free-field, for the observation points located at $6\,\mathrm{m}$ and $12\,\mathrm{m}$ from the track. A DVA on vehicle is only analysed, since, for the same mass, a DVA on track is less advantageous. Two DVA masses are analysed: $m_a = 100\,\mathrm{kg}$ and $m_a = 200\,\mathrm{kg}$. The impact of leading bogie is studied in order to easily observe the effect of the DVA (front and rear leading car induce the same ground vibration and thus only the front leading car is analysed). From this, it is shown that the DVA has a positive effect by modifying the ground vibration velocity shape and level. The amplitude reduction is around $20\,\%$ with a DVA. However, there is a small difference between the two DVA mass cases, showing that increasing the DVA mass does not always result in greater amplitude reduction. Corresponding one-third octave band analysis (Figure 8) confirms this observation by showing an effective reduction of the vibration magnitude around not only the target frequency, but also (to a lesser extent) at higher frequencies (as observed in the FRF results). Also, no notable difference between the two DVA masses is observed. Similar observations can be done for the central bogie and additional findings can be found in [18].



(a) $m_a = 10\,\mathrm{kg}$ (b) $m_a = 20\,\mathrm{kg}$ (c) $m_a = 30\,\mathrm{kg}$

(d) $m_a = 40\,\mathrm{kg}$ (e) $m_a = 50\,\mathrm{kg}$ (f) $m_a = 60\,\mathrm{kg}$

(g) $m_a = 70\,\mathrm{kg}$ (h) $m_a = 80\,\mathrm{kg}$ (i) $m_a = 90\,\mathrm{kg}$

(j) $m_a = 100\,\mathrm{kg}$ (k) $m_a = 200\,\mathrm{kg}$ (l) $m_a = 500\,\mathrm{kg}$

Figure 5: Frequency response functions at the motorised wheel (driving point) with an excitation originated by the leading car for DVA placed on the vehicle

5. Conclusion

A design of DVAs was carried out in order to evaluate the potential of such mitigation measures in the case of urban conditions. The ground vibrations induced by a localized defect was retained in order to calibrate such a device. The modal decomposition theory was applied to the vehicle/track model to

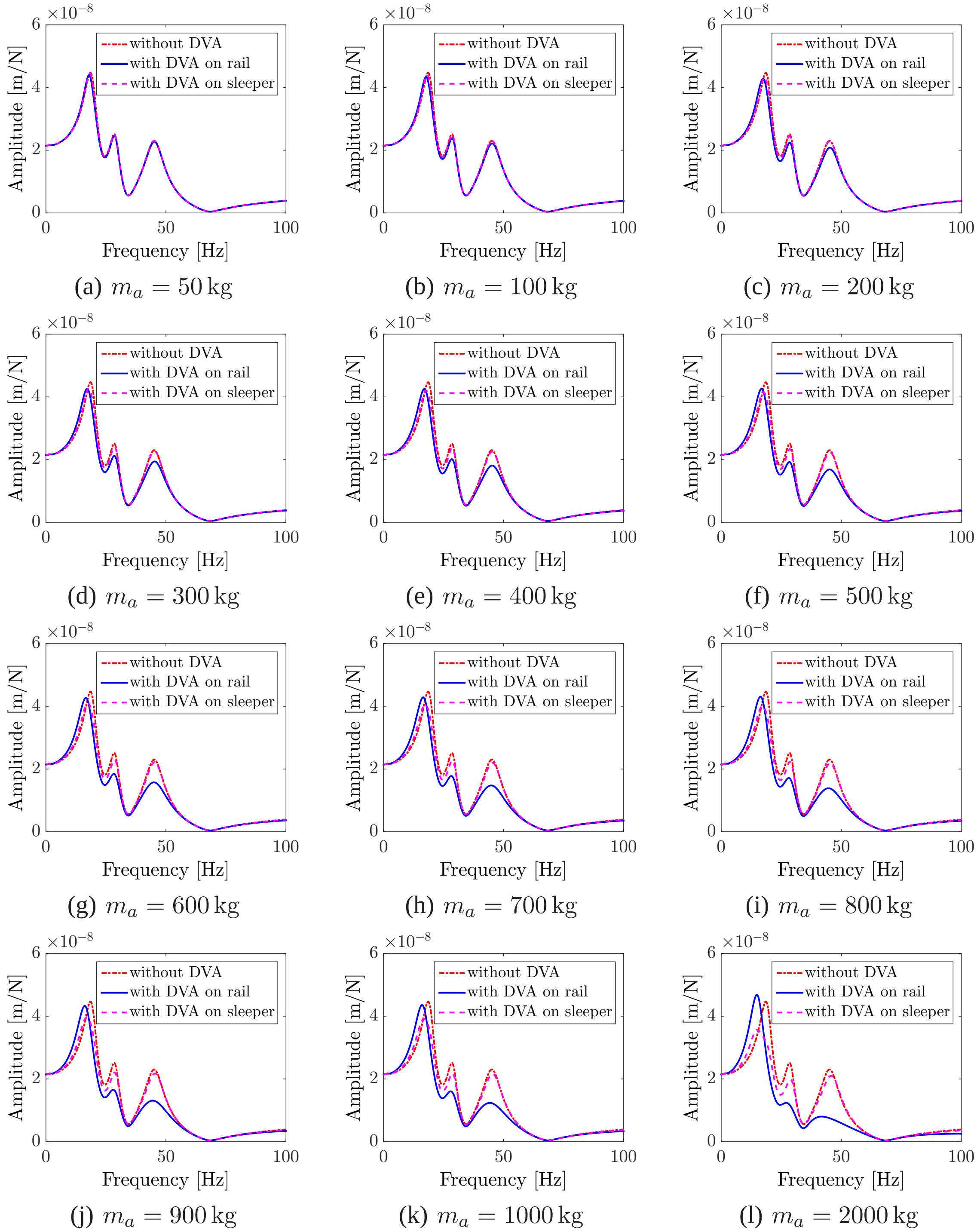
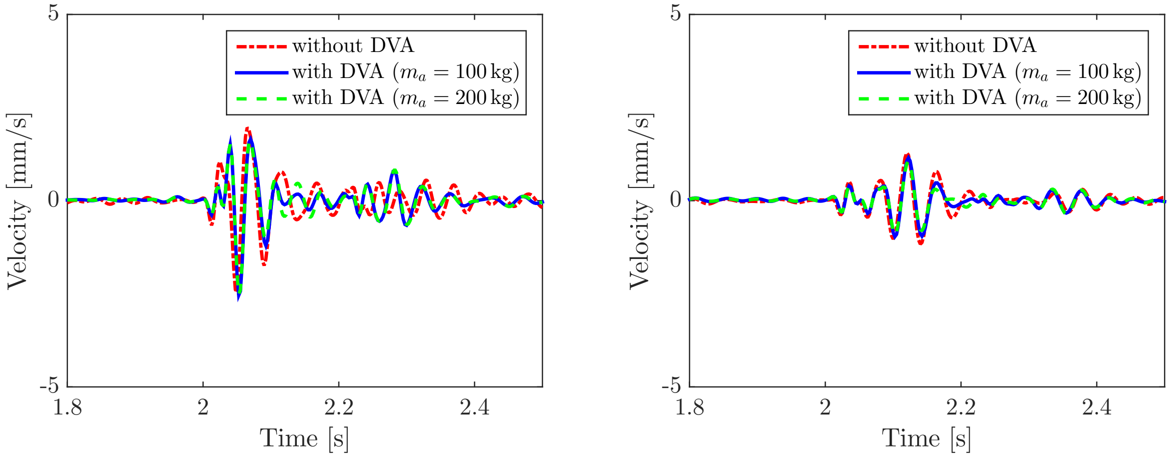


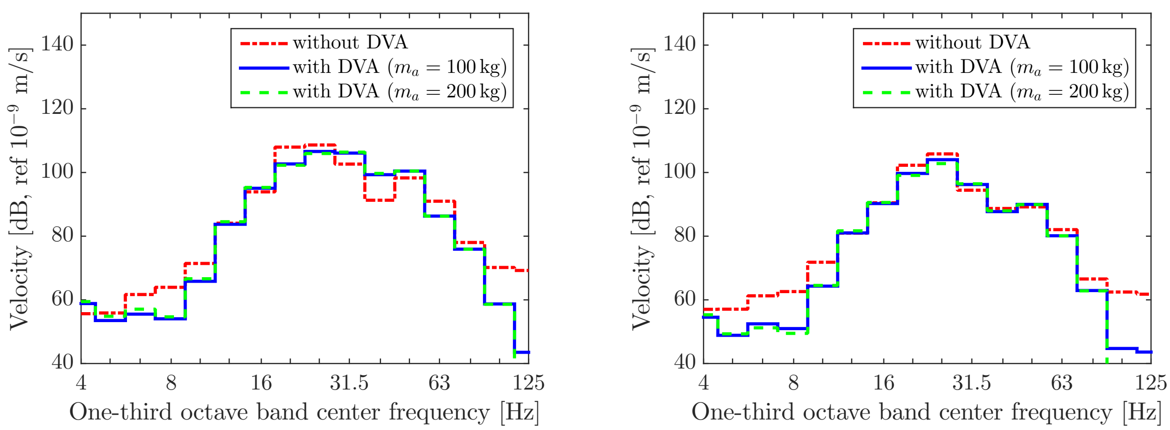
Figure 6: Frequency response functions at the motorised wheel (driving point) with an excitation originated by the leading car for DVA placed on the track.

obtain the optimal DVA parameter values for any location. Various simulations were performed on FRFs and it was observed out that a DVA placed on the vehicle, close to the excitation contributor, is more efficient than DVAs located on the track. Additional investigations, including a complete analysis of the vehicle/track/soil model, confirm the last finding and provide a significant gain in terms of ground vibration level reduction when the DVA is placed on the vehicle.



(a) at $6\,\mathrm{m}$ (b) at $12\,\mathrm{m}$

Figure 7: Predicted vibration velocity at $6\,\mathrm{m}$ and $12\,\mathrm{m}$ from the track due to the T2000 tram passing over a singular defect at a speed of $30\,\mathrm{km/h}$: time history of the vibration induced by the leading car.



(a) at $6\,\mathrm{m}$ (b) at $12\,\mathrm{m}$

Figure 8: Predicted vibration velocity at $6\,\mathrm{m}$ and $12\,\mathrm{m}$ from the track due to the T2000 tram passing over a singular defect at a speed of $30\,\mathrm{km/h}$: frequency analysis of Fig. 7.

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