# **Train-Induced Vibrations on Urban Metro and Tram Turnouts**

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**Abstract:** In contrast to standard track, where train-induced vibrations are mainly related to track irregularities, the dynamic response of turnouts is dominated by the nonuniform geometry of wheel-rail contact and variations in track flexibility. Such peculiarities are responsible for the development of strong vibrations during train passage. At the point of rail intersection (heart), where a gap is unavoidable to provide the necessary wheel flange clearance, the system is subjected to severe impact loading, critical for the design and maintenance of railway tracks. Especially in the case of urban turnouts, the vibration levels are also directly related to the exerted noise nuisance. This paper presents two analysis methods to simulate train-turnout interaction. The first is based on a multibody model of the trainset and of wheel-rail contact, utilizing a simplified finite element model for the turnout. The second focuses on the details of the turnout, which is modeled with three-dimensional finite elements, utilizing a simplified model to compute impact loading due to wheel passage over the flange-way gap. The two models are validated against line measurements on three different urban metro and tram networks. A parametric analysis is conducted to investigate the role of soil-structure interaction, which is shown be important for the dynamic response of the system.

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## Introduction

Turnouts (Fig. 1) are used to allow two rail tracks to intersect at the same level (Esveld 1989). As shown in Fig. 2, a standard turnout consists of three main parts: (1) the switch blades movable parts used to control the direction of train passage; (2) the heart of turnout (or crossing)—the central part where the two rail tracks intersect; and (3) the closure rail—a section that is necessary to connect the switch blades with the crossing. Evidently, due to the unavoidable existence of this variety of components, the system is characterized by sudden variations in track flexibility. Most importantly, since the wheels of a train vehicle have to roll over different components, the geometry of wheel-rail contact is rather nonuniform. At the point of rail intersection, where a flange-way gap is necessary to provide wheel flange clearance, the change in wheel-rail contact is quite sudden, lead-

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ing to impacts and "jumps" of the wheels. Impacts may also occur at the switch due to the shape and flexibility of the blades. Hence, in contrast to standard (straight) track superstructure, where traininduced vibrations are mainly related to track irregularities (Esveld 1989; Bode et al. 2000; Giannakos 2000; Kaynia et al. 2000), the vibratory response of turnouts is dominated by impacts.

Such impacts are responsible for the short service life of turnouts, and the increase of the associated maintenance cost. Given that these systems constitute the most expensive single items of a train track, their contribution to the overall maintenance of a rail track is rather substantial. Especially in the case of urban metro or tram turnouts, turnouts usually constitute the main source of noise nuisance, which is directly related to public acceptance of new or existing urban rail networks. The impact-induced vibrations constitute the source of such noise disturbances. This paper is based on the work performed in the EU-funded research project "TURNOUTS," aiming to reduce noise, vibration, and maintenance costs of turnouts, and allowing for improvement of existing urban rail networks through development of new concepts.

Before proceeding to the development of new improved turnouts (Anastasopoulos et al. 2009), it is necessary to understand the behavior of existing systems. To this end, two different simulation methods of train-turnout interaction have been developed and are presented herein. The first method (developed by Politecnico di Milano), is based on a refined multibody model of the vehicle in combination with an accurate model of wheel-rail contact, whereas for the turnout a simplified finite-element (FE) model is employed. The second method (developed by the National Technical University of Athens), employs a detailed threedimensional (3D) FE model of the turnout, also taking account of soil-structure interaction, whereas a simplified model is developed to compute the impact loading due to wheel passage. The two methods are complementary to each other: the first focuses



Fig. 1. Photo of crossing of typical railway turnout

on the details of wheel-track contact, providing a robust method to take account of the peculiar geometry of the system; the second focuses on the turnout structure, providing insight to the mechanisms of wave propagation through the structure and the surrounding soil.

The two modeling methods are validated against line measurements on three reference turnouts: (1) two tram turnouts of STIB in Brussels and De Lijn in Antwerp; and (2) a metro turnout of RATP in Paris. The three reference turnouts are characterized by different service conditions, regarding the train speed, the type of the rolling stock, the axle loads, and the level of rail wear. For both turnout types (tram and metro), a sensitivity analysis is conducted to highlight the effect of soil-structure interaction on the dynamic response of the system.

## Method A: Multibody Model

Following the multibody formulation of Shabana (1989), the trainset is decomposed into different modules, representing car bodies and bogies. For each module, the equations of motion are written with respect to a local moving frame traveling along the ideal path of the module, defined by the geometry of the line. The equations of the trainset are linearized (with respect to kinematic nonlinear effects only), assuming the motion to be a small perturbation around the large motion of the moving reference. For car bodies and bogie frames, a rigid body motion with constant forward speed is assumed, introducing five degrees of freedom per body. A flexible body description, based on modal superposition (Diana et al. 1998) is introduced for the wheelsets.

The turnout (rails and sleepers) is modeled with Euler-Bernoulli beam elements. Besides from the main components of the turnout (switch panel, crossing, and closure panel) two sections of standard track before and after the turnout are also incorporated in the model to establish the appropriate boundary







**Fig. 3.** Schematic illustration of multibody modeling approach (Method A)

conditions. Spring-damper elements are utilized to model rail fasteners, while the ballast is simulated with discrete lumped masses (one below each sleeper). Track foundation flexibility is incorporated in the model through an equivalent beam resting on a viscoelastic layer.

The overall logic of the model is schematically illustrated in Fig. 3. Equations of motion are written separately for train and track

$$[M_T]\ddot{\mathbf{X}}_T + [C_T]\dot{\mathbf{X}}_T + [K_T]\mathbf{X}_T = \mathbf{F}_{\mathrm{TC}}(\mathbf{X}_T, \dot{\mathbf{X}}_T, \mathbf{X}_V, \dot{\mathbf{X}}_V)$$
(1)

$$[M_V]\ddot{\mathbf{X}}_V + [C_V]\dot{\mathbf{X}}_V + [K_V]\mathbf{X}_V = \mathbf{F}_{\mathrm{IN}}(\mathbf{X}_V, \dot{\mathbf{X}}_V) + \mathbf{F}_{\mathrm{INT}}(\mathbf{X}_V, \dot{\mathbf{X}}_V) + \mathbf{F}_{\mathrm{VC}}(\mathbf{X}_T, \dot{\mathbf{X}}_T, \mathbf{X}_V, \dot{\mathbf{X}}_V)$$
(2)

where  $\mathbf{X}_T$ =vector of turnout nodal coordinates;  $[M_T]$ ,  $[C_T]$ , and  $[K_T]$  represent the mass, damping, and stiffness matrices of the turnout;  $\mathbf{F}_{TC}$ =vector of generalized nodal forces on the turnout, corresponding to wheel-rail contact forces;  $\mathbf{X}_V$ =trainset coordinates;  $[M_V]$ ,  $[C_V]$ , and  $[K_V]$ =mass, damping, and stiffness matrices of the trainset;  $\mathbf{F}_{IN}$ =vector of inertial forces due to the non-inertial motion of the local moving frames (taken as reference for the trainset modules);  $\mathbf{F}_{INT}$ =vector of internal forces associated to the differential motion of the two local frames and to nonlinear internal forces that cannot be accommodated within the linear expressions of the left hand side of Eq. (2); and  $\mathbf{F}_{VC}$ =vector of generalized forces produced on the trainset by wheel-rail contact forces.

Due to the nonlinearity of the problem (associated to wheelrail contact, and to the existence of nonlinear elements in vehicle suspension), the problem is solved in the time domain. Since wheel-rail contact forces act as coupling terms, Eqs. (1) and (2) must be solved simultaneously: an iterative correction is introduced in the time step using Newmark's implicit scheme as modified by Argyris and Mlejnek (1991).

In stark contrast to standard track superstructure, where wheelrail negotiation is attained through single contact, the formation of multiple contacts between each wheel, and the different rails in the turnout is also probable. In addition, due to the spatial variation of rail profiles along the turnout, the number of potential contacts and the associated contact parameters (contact angles, local rolling radius, and profile curvatures) are constantly changing, not only with lateral, but also with the longitudinal position of the wheel along the track. To overcome this problem, a multi-Hertzian approach (Piotrowski and Chollet 2005) is utilized to define wheel-rail contact forces. For each time step, and for each wheel, a number of "potential" contact points is defined, based on the local wheel-rail geometry. A Hertzian contact problem is solved for every potential contact point to derive the normal contact force (for the points where contact is not attained, the contact force is equal to 0). Then, using the formulae of Shen et al. (1983) tangential creep forces are computed, based on the derived normal contact forces and of longitudinal and transversal creepages. For the contacts that are found to be active in each wheel, the computed contact forces are added together; they are then transformed into vectors  $\mathbf{F}_{TC}$  and  $\mathbf{F}_{VC}$  through application of the virtual work principle (for more details see Braghin et al. 2006).

A further complication arises from the peculiar threedimensional geometry of the turnout: (1) as the wheel passes from the stock rail to the frog nose, the sudden change of wheel-rail contact is responsible for sudden vertical wheel movement ("jump" and impact); and (2) when the wheel is transferred from one rail to another (on the switch blade and on the frog nose), the dynamic response is complicated due to the difference in vertical and lateral stiffness. The first complication is treated through introduction of a generalized description of the contact plane, replacing the contact angle parameter with a two-component rotation vector, which allows description of the general inclination of the plane tangent to wheel-rail contact. To cope with the second complication, different potential contacts occurring on the same wheel are associated with different beam elements, representing the situation of a single wheel contacting more than one rails (of different inertia and stiffness).

## Method B: 3D Finite-Element Model

The second method focuses on the turnout structure, which is modelled using 3D finite elements, utilizing the FE code ABAQUS (2004). All turnout components (rails, crossing nose, sleepers, railpads, etc.) are modeled through hexahedral bricktype elements. Besides form the central part of the turnout, which is modeled in 3D, a composite boundary is introduced at the two ends to incorporate the effect of rail continuation. The boundary consists of beam elements to model the neighboring rails and sleepers, and springs-dashpot elements for the ballast. The springs represent the compliancy of the ballast, while the dashpots capture the radiation damping through ballast and subsoil. This way, waves propagating through the rails are allowed to radiate through the boundaries, not getting unrealistically trapped within the model.

A simplified analytical procedure is developed to compute the loading to the turnout, focusing on wheel impact at the area of the flange-way gap, which has been shown to constitute the main source of dynamic distress of a turnout (Anastasopoulos and Gazetas 2007). All other wheel-track interaction phenomena are not considered. If the geometry of the running surface of the turnout and of the wheels of the vehicles were perfect, such passage would rather be of a smooth transition. However, when the wheels are worn (and therefore their geometry is not ideal), then wheel passage over the flange-way gap is dominated by the aforementioned impact. The perfect shape assumption would tend to be realistic only for brand new vehicles (or for recently rehabilitated wheels). However, even in such a case, perfect contact would be realistic only when the turnout is also brand new (or just after it has been refurbished). Hence, the perfect contact assumption will only be valid for short time periods, during which both the turnout and the wheels are perfect.

As schematically illustrated in Fig. 4, when the wheel (of mass m) passes over the flange-way gap it lifts off at Point "A," follows an accelerating movement due to the compressed (by the weight



**Fig. 4.** Flange-way gap constitutes main source of dynamic distress of turnout. When wheel (of mass *m*) passes over this gap it lifts off at point "A," follows an accelerating movement due to compressed (by weight of car *Mg*) primary suspension spring (*k*), and eventually impacts ramp at point "B." Vertical impact velocity  $V_z$  depends on geometry of ramp (gap  $\delta$ , length *L*), properties of vehicle (*m*, *M*, and *k*), and horizontal train velocity  $V_V$ .

of the car Mg) primary suspension spring (k), and eventually impacts the ramp at Point "B." The vertical impact velocity of the wheel  $V_z$  depends on the geometry of the ramp (gap  $\delta$ , length L), the properties of the vehicle (m, M, and k), and the horizontal train passage velocity  $V_V$ .

At time t=0, the wheel "jumps" off the supporting rail with a horizontal velocity u. From this point on, the forces acting on the wheel are the gravitational (M+m) g and the spring reaction k  $(y-y_M)$ , where  $y_M$  is the initial compression of the primary suspension spring. The motion of the wheel is composed by two independent components: a constant-velocity motion in the horizontal direction, and an oscillatory motion in the vertical direction (the damping ratio  $\xi$  of the primary suspension is neglected)

$$x = V_V t \tag{3}$$

$$y = \frac{(M+m)g}{k} [1 - \cos(\omega t)]$$
(4)

Eliminating time in the above equations of motion, we compute the wheel orbit. Considering an idealized (planar) surface for the rail, the displacement vector of the impact point  $(x_i, y_i)$  corresponds to the solution of the following system:

$$y = \frac{(M+m)g}{k} \left[ 1 - \cos\left(\omega \frac{x}{u}\right) \right]$$
(5)





**Fig. 5.** Tram turnout of DeLijn: (a) photo of turnout; (b) 3D finiteelement model (Method B)



**Fig. 6.** Eigenfrequency analysis (Method B) of tram turnout of DELIJN— $k_{ballast}$ =35 MN/m<sup>3</sup>,  $\xi_{ballast}$ =15%: vertical displacement contours for Modes 1 and 2

$$y = d - \frac{d}{L}x\tag{6}$$

where d=function of the wheel wear w; rail wear  $w_R$ ; and the flange-way gap  $\delta$ ; and L=length of the ramp. This way, for a given horizontal train passage velocity  $V_V$ , the wheel velocity and the point of impact can be computed. Hence, the loading to the turnout is an impact velocity and not a contact force.

# Tram Turnouts of STIB and DELIJN

Two tram turnouts were selected as a reference. The first one, from the STIB tramway network in Brussels, is a turnout with grooved type rail. It was tested under the passage of an articulated trainset at a maximum speed of 15 km/h. The second, from the tram network of De Lijn in Antwerp [Fig. 5(a)], is also a grooved rail type turnout, but is equipped with a "flange bearing" crossing: i.e., when the wheel passes over the flange-way gap, it is temporarily supported through its flange. Trainset (articulated) and negotiation speed (20 km/h) were rather similar. Since the two turnouts are quite similar, and to be consistent with space limitations, we confine our discussion to the latter case (De Lijn). First, we present characteristic analysis results to gain some insight in the response of the system. Then, we compare our analytical predictions (of both methods) to line measurements in order to validate their effectiveness. Finally, a short sensitivity study is presented, highlighting the effect of soil-structure interaction.

## Analysis Results

The FE model of the De Lijn turnout is illustrated in Fig. 5(b). Sleepers, rails, and heart are modeled with hexahedral brick-type elements, while the supporting ballast and subsoil is modeled with spring-dashpot elements. An eigenfrequency analysis is performed to explore the dynamic response of the system. The aim is to extract the dominant vibration mode shapes of the system, and its dominant natural frequencies. It is noted that this kind of analysis can only be elastic. This means that the effect of debonding and separation-uplift that may occur between the sleepers and the ballast is not incorporated in the model. In other words, it is tacitly assumed that the ballast can sustain tensile forces. This is obviously an unrealistic simplification, however, since the ballast is originally in compression, due to the self-weight of the system, tension will only occur if the tensile forces (upward direction) manage to overcome the original compression. Furthermore, since the sleepers are not resting on top of the ballast, but are practically embedded in it, such a situation is not very easy to occur. Therefore, the elastic assumption for the ballast can be held as a reasonable first approximation.

Typical analysis results (for the case of "soft" ballast,  $k_{\text{ballast}} = 35 \text{ MN/m}^3$ ) are presented in Fig. 6, in terms of vertical displacement contours for the first two mode shapes. In the first—and dominant—mode (42 Hz), the heart of the turnout is moving mainly in the vertical direction (upwards) and bending longitudinally, causing transverse bending of the supporting sleepers and lateral opening of the rails. The second mode (49 Hz) is dominated by transverse movement of the heart. The first mode was



**Fig. 7.** Comparison of time history of vertical crossing nose acceleration for DeLijn turnout

found to compare well with the results of modal analysis of impedance measurements conducted by D2S, both in terms of shape and frequency: the measurements showed the first mode to be at 44 Hz, instead of 42 Hz of the analysis. Since the assumption of  $k_{\text{ballast}}$ =35 MN/m<sup>3</sup> was rather crude (mainly based on experience), this result is considered rather positive.

### Validation against Line Measurements

To assess the accuracy and reliability of the two simulation methods, numerical results are compared with line measurements. For this purpose, dynamic time history analysis is conducted using both methods. Among the available measurements, rail acceleration at the frog nose (very close to the point of impact) is taken as a representative term of comparison of the two methods with experimental data. Since Method B only considers the vertical excitation due to wheel impact, the comparisons is confined to the vertical acceleration. The time histories of computed *and* measured vertical accelerations are low-pass filtered with a cutoff frequency of 500 Hz, which is actually the limit of validity of the two methods (due to element size).

A comparison of the low-pass filtered acceleration time histories is shown in Fig. 7 for a 100 ms time frame, centered over the passage of a wheel over the frog. The wheel passage producing the highest acceleration peak was selected for comparison. The analytical prediction of Method A is in fairly good agreement with the measurements: the model captures the peculiar type of impact excitation produced by the flange bearing crossing type. The passage of the wheel over the crossing panel is "smoothed" by gradually decreasing the depth of the rail groove, until contact takes place between the outer surface of the wheel flange and the bottom of the rail groove, so that the wheel tread is then lifted until completion of the passage over the crossing. This way, instead of a large impact on the crossing nose, two impacts of smaller magnitude are produced: (1) when contact is transferred from the wheel tread to the flange tape; and (2) when flange contact is restored after the passage of the crossing.

The simulation using Method B is also in reasonably good agreement with the measurements. However, since this method simulates only one impact (in a rather simplified manner), the numerically predicted time history refers to the impact of the outer flange surface on the bottom of the rail groove that takes place during the entrance of the wheel into the crossing panel. Although the duration of the acceleration pulses is quite different, the maximum values are in very good agreement with the measurement.

Table 1 synopsizes the results of the comparison in terms of maximum and minimum peak values, RMS, and difference between measurement and analysis results in dB. In terms of maximum and minimum acceleration, Method A achieves the best results: the difference from the measurement ranges from 0.2 to 0.5 dB, instead of -0.7 to -1.3 dB of Method B. On the other hand, despite the crude modeling of wheel impact Method B is better in terms of RMS performance (a difference of 0.4 dB instead of 1.9 dB of Method A). Overall, the two methods provide comparable results in terms of vertical acceleration on the crossing nose, and are both in very good agreement with the measurements.

## Effect of Soil-Structure Interaction

To illustrate the effect of soil-structure interaction on the dynamic response of the system, we parametrically vary the stiffness of the ballast (i.e., the soil) from  $k_{\text{ballast}}=35 \text{ MN/m}^3$  to 70 and 100 MN/m<sup>3</sup>. A first conclusion is that the mode shapes of the system are practically insensitive to ballast stiffness. In stark contrast, as shown in Table 2, the eigenfrequency of each mode is altered substantially: an increase of  $k_{\text{ballast}}$  from 35 to 100 MN/m<sup>3</sup> leads to an increase of the dominant frequency of the system from 42 to 63 Hz. As it will be shown in the sequel (for the metro

Table 1. Comparison of Maximum, Minimum, and RMS Values of Vertical Crossing Nose Acceleration for the Tram Turnout of De Lijn

	Measurement (g)	Method A $(g)$	Measurement versus Method A (dB)	Method B $(g)$	Measurement versus Method B (dB)
Maximum amplitude	7.17	7.32	0.2	6.20	-1.3
Minimum amplitude	6.75	7.12	0.5	6.20	-0.7
RMS	1.33	1.65	1.9	1.39	0.4

**Table 2.** Effect of Soil-Structure Interaction: Eigenfrequencies (for First

 Five Modes) of Tram Turnout of De Lijn with respect to Ballast Stiffness

Mode	$k = 35 \text{ MN}/\text{m}^3$	$k=70 \text{ MN}/\text{m}^3$	$k = 100 \text{ MN}/\text{m}^3$
1 (Hz)	41.6	47.8	62.6
2 (Hz)	49.3	55.7	71.0
3 (Hz)	49.5	55.8	71.8
4 (Hz)	53.0	60.9	79.8
5 (Hz)	54.9	62.2	80.2

turnout of RATP), such a difference in the stiffness of the system can play an important role for its vibratory response.

# Metro Turnout of RATP

Situated at the underground network of RATP in Paris, this is a typical turnout equipped with vignole-type rails. The trainset is formed by four-axle coaches with bogies, traveling at speeds of up to 55 km/h. As for the previous case, we first present characteristic analysis results to provide some insight into the response of the system. Then, analytical predictions of the two methods are compared to line measurements, and finally, the results of a short sensitivity study on the effect of soil-structure interaction are presented.

### Analysis Results

As illustrated in Fig. 8, the FE model (Method B) of the turnout comprises sleepers, rails and guardrails, heart, and baseplates. As for the De Lijn turnout, an eigenfrequency analysis is first performed to explore the dynamic response of the system. Typical analysis results (for  $k_{\text{ballast}}=35 \text{ MN/m}^3$ ) are shown in Fig. 9, in terms of vertical displacement contours for the first two mode shapes. The first mode [Fig. 9(a)] is very similar to that of the De Lijn turnout, with the heart of the turnout moving upwards, causing transverse bending of the sleepers and opening of the rails. The second mode [Fig. 9(b)] is dominated by bending of the heart, which is now moving upward at the back of the turnout and downward at the front.

Fig. 10 depicts six snapshots of FE deformed mesh (Method B) for a dynamic time history analysis of the turnout subjected to wheel impact. The wheel impact the heart of the turnout at time t=3 ms, which starts moving downward. At t=5 ms, the heart of the turnout is still moving downward, reaching its ultimate settlement for t=8 ms. Then, it starts moving upward, reaching its maximum uplift at t=23 ms. The impact of the wheel at the heart



**Fig. 8.** 3D finite-element model of RATP turnout (Method B)



**Fig. 9.** Eigenfrequency analysis (Method B) of metro turnout of RATP— $k_{\text{kballast}}$ =35 MN/m,  $\xi_{\text{ballast}}$ =15%: vertical displacement contours for Modes 1 and 2

of the turnout generates substantial stressing, as illustrated in Fig. 11 (contours of Mises stresses). Observe that the area of large stress concentration (dark area in the FE figure) coincides with the actual area of increased wear of the turnout (shiny area at the photo).



Fig. 10. Snapshots of FE deformed mesh (Method B) for RATP turnout



**Fig. 11.** Area of wheel impact: comparison of actual turnout (RATP) with contours of Mises stress as predicted through FE analysis (Method B)

#### Validation against Line Measurements

The comparison between analysis and measurement is shown in Fig. 12, in terms of low pass filtered time history of vertical rail acceleration on the frog nose. Observe that a main peak occurs



**Fig. 12.** Comparison of time history of vertical crossing nose acceleration for the RATP turnout

when the wheel is transferred from the stock rail to the frog nose, which is preceded by two smaller peaks. These earlier peaks are attributed to concentrated irregularities over the railhead, before and after the frog nose, as observed during visual inspection of the turnout. Finally, a relatively low but not negligible level of "random" vibration can also be observed over the whole duration of the signal, representing the effect of wide band random irregularity of the rails due to wear, alignment errors, etc. Such effects have not been considered in the analysis.

The numerical prediction using Method A shows a main peak during wheel passage over the crossing nose, with positive and negative extreme values in accord with the measurements. The shape of the maximum peak is symmetric, as in the measurement and the small discrepancies between measured and simulated extreme values may be explained by the fact that an equivalent worn geometry of the rails along the crossing panel had to be assumed (no such measurements were available). Method B also predicts a large acceleration peak during impact of the wheel over the turnout structure, representing the passage of the wheel over the crossing nose. The shape of this peak shows some asymmetry, the negative maximum amplitude being larger than the positive one, but the overall levels of vibration are well in line with the results of the measurements.

The results of the comparison in terms of maximum and minimum peak values, RMS, and difference between measurement and analysis results in dB are summarized in Table 3. Comparing the results to the tram turnout of De Lijn (Table 1), it is quite clear that the vibration levels are an order of magnitude higher. This is due: (1) to the substantially higher train speed (55 instead of 20 km/h); and (2) to the fact that this turnout was in quite poor geometric condition at the time of the measurement. While Method A tends to underestimate the maximum and minimum vertical acceleration on the crossing nose, Method B does the opposite (overestimation). In all cases, the difference from the measurements does not exceed 3 dB with respect to the maximum and minimum values. In terms of RMS values, both methods are very good agreement with the measurement: their difference ranges from 0.3 dB (Method B) to 0.4 dB (Method A). As for the De Lijn tram turnout, overall, the two methods provide comparable results and are in very good agreement with the measurements.

## Effect of Soil-Structure Interaction

We investigate the effect of soil-structure interaction varying the stiffness of the ballast from  $k_{\text{ballast}}=35 \text{ MN/m}^3$  to 70 and 100 MN/m<sup>3</sup>. First, we conduct an eigenfrequency analysis of the system. As for the De Lijn case, the mode shapes are practically insensitive to  $k_{\text{ballast}}$ , with the main difference lying in the frequency of each mode (Table 4).

To further investigate the role of ballast stiffens, we conduct a dynamic wheel impact time history analysis for the two extreme values:  $k_{\text{ballast}}=35$  and 100 MN/m<sup>3</sup>. The role of radiation damp-

Table 3.	Comparison	of Extreme a	and RMS	Values of	Vertical	Crossing 1	Nose	Acceleration	for RA	ATP T	urnout
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	Measurement (g)	Method A (g)	Measurement versus Method A (dB)	Method B $(g)$	Measurement versus Method B (dB)
Maximum amplitude	44.9	31.6	-3.0	50.3	1
Minimum amplitude	43.8	37.6	-1.3	65.2	3.5
RMS	8.46	8.9	0.4	8.7	0.3

**Table 4.** Effect of Soil-Structure Interaction: Eigenfrequencies (for First

 Five Modes) of Metro Turnout of RATP with respect to Ballast Stiffness

Mode	$k = 35 \text{ MN}/\text{m}^3$	$k=70 \text{ MN}/\text{m}^3$	$k = 100 \text{ MN}/\text{m}^3$
1 (Hz)	42.2	55.3	63.6
2 (Hz)	52.5	67.3	75.6
3 (Hz)	53.9	68.0	76.9
4 (Hz)	56.0	73.8	85.1
5 (Hz)	56.5	74.4	85.8

ing, provided by the underlying subsoil, is also investigated. Two values of damping are investigated: 15 and 30%. Fig. 13 compares the vertical acceleration and displacement time histories at the heart of the turnout (close to the point of impact) for the four cases investigated. The increase of ballast stiffness leads to a decrease of the vertical acceleration  $a_z$  and of the displacement  $\Delta_z$ . As expected, the increase of radiation damping (from 15 to

30%) also has a positive effect: a decrease of the maximum amplitude of  $a_z$  and  $\Delta_z$ , accompanied by faster attenuation of the motion. As depicted in Fig. 14, the conclusions are qualitatively similar in the case of the rails, with the differences being more pronounced.

## Conclusions

This paper has presented some of the results of a research project dealing with the reduction of impacts and vibration at urban railway turnouts.

Two alternative complementary methods have been developed to simulate the dynamic response of turnouts. The two methods were validated against line measurements on three reference turnouts, two form the tramway networks of STIB and De Lijn, and one from the metro network of RATP. The results of these com-



Fig. 13. Effect of soil-structure interaction—dynamic analysis (Method B) of metro turnout of RATP: vertical acceleration and displacement time histories at heart (close to point of impact)



Fig. 14. Effect of soil-structure interaction—dynamic analysis (Method B) of metro turnout of RATP: vertical acceleration and displacement time histories at rails

parisons are quite satisfactory, since the two methods are in good agreement and the numerical results match the measurements well. It is emphasized that the three validation examples represent a wide range of possible train-turnout conditions of urban rail transportation.

A short parametric study on the effect of soil-structure interaction has also been presented. It is shown that the increase of ballast or subsoil stiffness, as well as the increase of radiation damping, tend to ameliorate the dynamic response of the turnout: accelerations and displacements are decreased noticeably. Interestingly, those effects are more pronounced for the adjacent rails rather than for the heart of the turnout. The latter is directly affected by the impact and the influence of the soil (ballast) is not dominant. In contrast, the rails are only affected indirectly: the impact-generated waves have to pass though the sleepers to actuate the rails, and hence the foundation reasonably plays an increased role.

The two modeling approaches described in this paper were used for the assessment of new turnout concept solutions aiming to reduce noise, vibration, and maintenance costs of turnouts, allowing for improvement of existing urban rail networks. Taking into account that turnouts usually constitute the main source of noise nuisance, which is directly related to public acceptance of new or existing urban rail networks, the importance of such ameliorations becomes clear. The results from this research activity are described in detail in Anastasopoulos et al. (2009).

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