REDUCTION OF RAILWAY INDUCED VIBRATION WITH SOFT FASTENING SYSTEMS FOR BALLASTED TRACKS

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Within the scope of the European Project RIVAS (Railway Induced Vibration Abatement Solutions), the vibration mitigation potential of soft fastening systems for ballasted track was investigated through numerical simulations, laboratory tests and real scale measurements. The results of these studies are summarized in this paper. For the numerical simulations, a 2.5 Dimensional methodology is used based on a coupled FE-BE method (the properties of the soil and the track are invariant along the axis of the track). From these simulations, it is shown that soft fastening systems have a low pass filter effect on ground vibration. At the resonance frequency for which the unsprung mass of the vehicle bounces on the track stiffness (around 60 Hz), ground vibration is amplified up to 15 dB. Above this resonance frequency, there is a cut-off effect leading to a mitigation of ground vibration up to 20 dB. For the laboratory measurements, a single sleeper equipped with two pieces of rail is laid in a ballast box. A static load is applied to the rails by a hydraulic jack as the dynamic excitation is provided by a modal shaker. A classical Fastclip system is tested with different rail pads as well as two innovative fastening systems produced by Pandrol: the Vanguard (rail held by its web) and the Vipa Valiant (baseplate and two resilient layers). The laboratory tests confirmed the performances expected from the simulations. Finally, the Vipa Valiant is tested in a commercial track, for three different configurations (different stiffness for the Vipa Valiant system obtained with different rail-pad/baseplate-pad combinations).

1. Introduction

The European project RIVAS (Railway Induced Vibration Abatement Solutions) aims for the reduction of the environmental impact of ground-borne vibration from rail traffic. This issue is assessed in a wide range of topics within the project, ranging from the generation mechanisms to the response of the buildings, and including the propagation path. Concerning the mitigation measures at the source, soft track designs appear to be cost effective solutions. Among the different track designs and track components involved in this problematic, this paper focuses on soft fastening systems for ballasted track in alignment.
Two original designs are proposed by Pandrol Ltd, member of the RIVAS consortium: the Vipa Valiant (previously called DFC Valiant) and the Vanguard. They were originally designed for slab tracks for which softer fastening systems are allowed compared to ballasted tracks. Within the RIVAS project, both systems were modified by Pandrol for ballasted track designs, and especially in order to fit a common sleeper, also designed within RIVAS (by SATEBA).

The Vipa Valiant (Fig. 1 left) is a fastening system with two resilient layers: a rubber pad (a) is placed between the sleeper and a cast-iron baseplate (b) on which the rail is mounted through a Fastclip-like system including a classic rail pad (c). A wide range of stiffness can be achieved with the Vipa Valiant by using different rail-pad/baseplate-pad combinations. In this paper three different assemblies are considered which vertical dynamic stiffness according to EN-13146:2012 [1] are 25, 36 and 75 MN/m. The Vanguard (Fig. 1 right) is a fastening system where the rail is supported under the head and in the web with rubber wedges (d) and cast-iron side brackets (e), leaving the foot of the rail suspended (f). The vertical dynamic stiffness of the Vanguard was measured at 6 MN/m.

![Figure 1. Overview of the tested systems: Vipa Valiant (left) and Vanguard (right).](image)

First, a numerical study is performed in order to appraise the performances of soft fastening system depending on other track and soil parameters. Using the software TRAFFIC c⃝ developed by KU Leuven, the ground vibration levels at several distances from the track are computed for a passenger train pass-by. In this software, a 2.5 dimensional model is used for the track, based on a coupled Finite Elements - Boundary Elements (FE-BE) modelling. This numerical model is presented in Section 2 as well as the Insertion Loss (IL) on ground vibration computed for soft systems compared to the standard one.

In Section 3, the Vipa Valiant, the Vanguard and classical Fastclip equipped with different rail pads are tested in laboratory. Each fastening system is tested with an artificial excitation applied to a complete assembly: two pieces of rail fastened to a single sleeper laid in a steel ballast box. In order to reproduce the specific loads experienced in real track conditions, two specific setups using a hydraulic jack and a modal shaker are used for the excitation. The results are assessed in terms of estimated stiffness and of ballast box vibration.

Finally, the Vipa Valiant system is tested in a commercial track. Using different rail pads and under baseplate pads, three Valiant configurations are tested successively on a track section of 100 m. Ground vibration levels are measured for a reference section and for the test section: the reference section is equipped with rail pads at 120 MN/m as the test section, which is crossed by the same trains, is equipped with one of the three Valiant systems (75 MN/m, 35 MN/m or 25 MN/m). The measurement setup and the results in terms of track receptance and IL on ground vibration at 8 m from the track are presented in Section 4.

2. Numerical simulations

When a ground vibration issue is raised for a railway track, whether it is new or not, soft designs are always considered as potential solutions. The real question is then to make a choice among the
different ways to achieve the track softening (ballast mats, under sleeper pads or floating slabs) but also among other existing measures (sheet piling walls, buildings insulation). The objective of the parametric study performed within the RIVAS project is to understand and predict the behaviour of a maximum of these mitigation measures depending on a large number of track and soil parameters.

The results presented in this Section concern the efficiency of soft fastening systems depending on their stiffness, on the sleeper type (mass, width and length), and on the soil conditions (number of layers and associated shear wave velocities). The model used for the numerical simulations is presented in Subsection 2.1 and the results are given in Subsection 2.2. The reader may refer to [2] for a complete description of the model.

2.1 Description of the model

The software TRAFFIC (C) developed by KU Leuven is used for these simulations [3, 4]. It is based on a coupled 2.5 dimensional FE-BE method for the track model which is invariant in the longitudinal dimension \( \vec{z} \) (see Fig. 2). The rails are modeled as Euler beams (a) and an equivalent continuous model is used for the support: a spring-damper layer for the rail pads (b) and a mass layer rigid in its cross-section for the sleepers (c). The ballast is modeled as an elastic continuum with 2.5D finite volume elements (d) and the soil is modeled by means of boundary elements at the interface between the ballast and the soil (e). The system has four degrees of freedom: the vertical displacement of the two rails and the sleeper, and the sleeper rotation around \( \vec{z} \).

The ballast (d) is 30 cm high with a density of 2000 kg/m\(^3\), a width of 5.6 m at the bottom and a width of 3.6 m at the top. The speed of the different waves in the ballast is 300 m/s for the shear wave and 600 m/s for the dilatational wave, with a damping of 2%. The sleeper layer (c) has a width of 2.6 m and a linear density of 541.6 kg/m. The stiffness of the resilient layer (b) is \( K_p/\alpha \) with a damping of 20%, where \( K_p \) is the rail pad stiffness and \( \alpha \) the sleeper spacing.

![Figure 2. Cross section of the FE-BE model used in the TRAFFIC (C) software.](image)

The model described in Fig. 2 is excited by vertical point forces representing the vertical wheel/rail interaction forces due to a regional train pass-by at speed \( v \). The only excitation mechanism considered here is the combine wheel/rail unevenness taken as a stationary Gaussian random process. The parametric excitation due to the periodicity of the rail support is ignored by the model itself (invariant along \( \vec{z} \)). The quasi-static excitation due to the moving loads is not considered, as it is negligible for ground-borne vibrations at conventional speeds [2]. The equations that link the unevenness Power Spectral Density (PSD) to the ground vibration PSD can be found in [2, 4].

2.2 Results

In this Section, the efficiency of soft fastening systems depending on the soil characteristics is investigated. The results are presented in terms of IL on ground vibration between a reference case and a test case. The rail pad stiffness is 150 MN/m for the reference case and 25 MN/m for the test case presented in this article.
The frequency of interest is the frequency at which the unsprung masses of the vehicle bounce on the track stiffness. The value of this frequency as well as the corresponding amplitude of the wheel/rail interaction force are mainly driven by the track stiffness and the wheelset mass. Thus, the rail pad stiffness, the sleeper type, the ballast height and the soil characteristics will not only influence the transmission of the vibration but also the generation of the excitation. With the parameters given in Subsection 2.1 and for a rail pad stiffness of 25 MN/m, the vehicle on track resonance frequency is equal to 40 Hz. The IL given in Fig. 3 with respect to frequency highlights the low-pass filtering effect provided by the track softening. Ground vibration are amplified up to 10 dB for \( f \leq f_{25} \), and then, after the cut-off frequency at around 60 Hz, an attenuation up to 20 dB is reached.

Fig. 3 also shows that the influence of the soil characteristics is low. The parameter study performed within the RIVAS project [2] showed a very low influence of other parameters. The sensitivity of under sleeper pads impact to extra parameters is similar except for the influence of the sleeper type which is important (weight and surface of contact with the ballast) [2].

The low frequency amplification is intrinsic to the model and inevitable according to it. However, previous measurements performed on tracks with under sleeper pads showed less or even no amplification for low frequencies [5]. This mitigation at low frequencies is related to the reduction of the parametric excitation [6]. This excitation generated by the stiffness variation within a sleeper bay is reduced for soft fastening systems. Depending on the value of the sleeper passing frequency compared to the vehicle on track resonance frequency, some effects can be compensated or amplified.

3. Laboratory tests

In order to confirm the results obtained from the numerical simulations, and also in order to test many different systems in controlled conditions, laboratory tests are performed. To be as representative as possible of the real track conditions, a ballast box is used. Within the RIVAS project, five different systems were tested: the Fastclip system associated with three different pads, the Vipa Valiant and the Vanguard. Their respective vertical dynamic stiffness according to EN 13146:2012 [1] are 120 MN/m, 80 MN/m, 37 MN/m, 25 MN/m (Valiant) and 6 MN/m (Vanguard). The experimental setup is presented in Subsection 3.1 and the results are given in Subsection 3.2.

3.1 Description of the experimental set-up

For each tested system, a sleeper is laid in the ballast box, with freshly poured ballast. The excitation is equally applied to two pieces of rail fastened to the sleeper. Two setups are used for the rail excitation: a small amplitude setup with a high frequency excitation of small amplitude under a static preload and a high amplitude setup with a low frequency excitation of high amplitude around a static component of high magnitude. A schematic representation of the setups is given in Fig. 4.
as well as the parameters of the excitations. The small amplitude setup should be representative of
the unevenness excitation which is broadband with relatively small amplitude considering the small
vertical displacement of the wheel/rail contacts. The amplitude of the excitation should be more close
to some kilo newtons but it was not reachable with our modal shaker. The high amplitude was used
as a try to be representative of the parametric excitation with a higher amplitude. In this case, the
frequency should be more close to some tens of newtons but it was not reachable with our hydraulic
jack.

Figure 4. Experimental set-up: small amplitude set-up (left) and high amplitude set-up (right).

3.2 Results

The following quantities are measured: the injected force, the relative displacement between
each rail and the sleeper, the vertical acceleration of the two rails, of the sleeper and of the ballast box
bottom. The original objective of these tests was to estimate the stiffness of the fastening systems in
realistic conditions and compared these results with those from the European standard EN 13146:2012
[1]. In this standard, the dynamic stiffness is estimated as the secant of an 18 kN – 68 kN cycle at 4
Hz in the Force/Displacement diagram.

With the small amplitude setup, the stiffness estimations for the different systems (not presented
here) showed very surprising results, be it in terms of values compared to normative measurements
results or in terms of behaviour with respect to frequency or preload. For example, the stiffness of
the standard SNCF railpad was found to vary nonlinearly between 1200 MN/m and 100 MN/m in the
[8-100] Hz frequency range (while expected at 120 MN/m according to [1]). It highlighted that the
stiffness characterization of a fastening system under realistic load conditions is a complicate issue,
from the understanding of the physical mechanisms to the capacity to reproduce them in a laboratory.
Therefore, for the small amplitude setup, we focus in this article on the vertical acceleration of the
ballast box which we suppose to be representative of the vibration transmitted to the ground. For
each system, the mobility between the injected force and the ballast box vibration is measured. The
resulting IL with the standard rail pad taken as a reference are given in Fig. 5 (left). Only few
negative IL are found except for the Vanguard which behaves close to what is expected from numerical
simulations (see Fig. 3 left). The mitigation is limited to 5 dB for the classical fastening systems that
use pads.

From the high amplitude setup, more consistent stiffness were measured. At 5 Hz under a 60
kN preload, the estimated stiffness are: 560 MN/m (SNCF standard railpad), 85 MN/m (Valiant) and
9 MN/m (Vanguard). These values are used to estimate the IL on ground vibration at 8 m with the
FE-BE method described in Section 2.1. The results are given in Fig. 5 (right) for the Vipa Valiant
and the Vanguard, the rail pad at 120 MN/m being the reference. Negative IL are found below 100
Hz for the Valiant and 40 Hz for the Vanguard. Above this cut-off frequency, positive IL are found up to 11 dB and 20 dB respectively.

![Image](image.png)

**Figure 5.** Performances of soft fastening systems from laboratory measurements. Left: IL on ground vibration at 8 m from numerical simulations using the stiffness measured with the small amplitude setup. Right: IL on the ballast box vertical vibration measured with the high amplitude setup.

For classical systems with rail pads, a ground vibration mitigation is possible only if a significant stiffness gap exists between the reference case and the mitigated case. The excitation mechanisms and the mechanical properties of the pads in real conditions seem to digress from what is used in the numerical simulations. As a consequence, the expected negative effects should be limited. Due to its original principle (load transmitted by shear instead of compression), the behaviour of the Vanguard system is closer to the predictions, with high IL found to the detriment of amplification effects at the vehicle on track resonance frequency.

4. **Full scale measurements in a commercial track**

In the last months of the RIVAS project, the mitigation measures that have been numerically studied until then were finally tested *in situ*. In order to implement new components in a commercial track, railway authorities require a drastic certification process, long and expensive. For these reasons, only the Vipa Valiant system was tested.

4.1 **Description of the field tests**

The test track is located in the north east of France in a freight corridor near the city of Florange. On the same track in alignment, two sites are equipped with sensors: a reference section and a test section. The reference section is equipped with rail pads of 120 MN/m, which is the regular track form for this line. For the test section which is 250 m away in the traffic direction, 100 m of track are equipped with the Vipa Valiant system. The measurement campaign was performed just after a partial track renewing (only the rail was kept). Using different combinations of rail pads and under baseplate pads, three Vipa Valiant assemblies were tested consecutively: Valiant 1 (75 MN/m according to EN 13146:2012), Valiant 2 (45 MN/m) and Valiant 3 (25 MN/m).

4.2 **Results**

The track dynamics are characterized with impact hammer measurements. For each Valiant configuration and for the reference section, the low frequency track receptance is measured with a large hammer at mid-span and the high frequency track receptance is measured at the same location.
with a smaller hammer. The results are given in Fig. 6, the thick lines indicating a coherence greater or equal to 0.9 between the injected force and the vibration signal.

In Fig. 6 (left), the rail on pad resonance frequency for unloaded track conditions is clearly seen for Valiant 2 and 3 around 160 Hz and 230 Hz respectively. The peak for Valiant 1 is less clear, between 80 Hz and 125 Hz. According to the phase shift (not shown here) and to Fig. 6 (right), we can state that the resonance occurs at 100 Hz. Together with the resonance at 570 Hz for the reference pad, it corresponds to stiffness of 14.2 MN/m, 36.4 MN/m, 75.2 MN/m and 462 MN/m for Valiant 1 to 3 and the standard pad respectively. In Fig. 6 (right), we can observe that above the rail on pad resonance frequency, the rail is decoupled from the track: the track receptance is close to the behaviour of a free rail, except at the pinned-pinned frequency. From these receptance measurements, we find stiffness in accordance with EN 13146:2012 values for Valiant 1 and 2 (see Subsection 4.1). However, the Valiant is found to be softer than expected and the standard pad stiffer (which is a common result).

For a given Valiant configuration tested on track, and for a given pass-by, the direct IL on Ground Vibration (GV) is defined as the difference in decibels between the GV levels at the reference section and the GV levels at the test section, integrated over the train pass-by, given in 1/3 octave band. For the direct IL, GV levels are compared on two different sites with the same excitation. For each Valiant configuration, around 80 trains pass-bys were recorded. The IL given in Fig. 7 are averaged over the trains which speed $V$ is between 60 and 70 km/h.

The relative IL is defined as the difference between the direct IL measured for the Valiant 1 (considered as a reference situation) and the direct IL measured for Valiant 2 and Valiant 3 (considered as a mitigated situation). In this case, the GV levels are compared on the same site, admittedly with excitations of different natures, but still brought back to equivalent excitation levels. Indeed, the GV levels at the test section are not compared directly between two different trains, but it is their respective direct IL that are compared.

In Fig. 7 (left), the behaviour expected from the numerical simulations and from the laboratory tests is retrieved: amplification of GV below the vehicle on track resonance frequency, attenuation above, and increasing effects with decreasing stiffness. However, some important differences can be noticed. First, the vehicle on track resonance frequency for Valiant 3 is found lower here compared to simulations or ballast box measurements (40 Hz instead of 63 Hz). This is in accordance with the stiffness estimated from the receptance measurements. Secondly, positive IL are found for low frequencies, especially in the case where the Valiant systems are compared together on the same track section. This is due to the parametric excitation which is mitigated in this case (the sleeper passing frequency is around 30 Hz).
Figure 7. Measured IL on ground vibration at 8 m for trains with speed $V$ between 60 and 70 km/h. Direct IL (left) and relative IL (right).

5. Conclusion

The work presented here summarizes the study performed within the RIVAS project concerning ground vibration mitigation using soft fastening systems for ballasted track. Starting with numerical simulations, it is expected that these kind of systems can provide up to 20 dB of attenuation on ground vibration for very soft systems (6 MN/m) compared to french standard rail pads (120 MN/m). This mitigation occurs above the resonance frequency of the vehicle bouncing on the track stiffness. Below and at this frequency, according to the numerical model and to the physics of spring-damper-mass systems, an amplification up to 10 dB is expected. During the full-scale measurements performed in a ballast box, the same tendencies were found with slight differences in the order of magnitude. This allowed to point out the difficulty of reproducing the load conditions that corresponds to the hypothesis of the simulations and/or to the dynamics of the wheel/track interaction.

Finally, the Pandrol Valiant system made of a baseplate and two resilient layers was tested in a commercial track. Using different materials for the rail pad and the under baseplate pad, three configurations for the Valiant were tested successively. Compared to standard rail pads with a stiffness of 120 MN/m, the softest configuration of the Valiant (25 MN/m) leads to a mitigation up to 10 dB after 50 Hz, with limited amplification for low frequencies. When the substructure influence is removed, even positive IL are found at low frequencies, and especially around the sleeper passing frequency. This indicates that the parametric excitation might be mitigated by soft fastening systems.

REFERENCES