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A double tuned rail damper—increased damping at the two first pinned–pinned frequencies

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Abstract

Railway-induced vibrations are a growing matter of environmental concern. The rapid development of transportation, the increase of vehicle speeds and vehicle weights have resulted in higher vibration levels. In the meantime vibrations that were tolerated in the past are now considered to be a nuisance. Numerous solutions have been proposed to remedy these problems. The majority only acts on a specific part of the dynamic behaviour of the track. This paper presents a possible solution to reduce the noise generated by the 'pinned–pinned' frequencies. Pinned–pinned frequencies correspond with standing waves whose nodes are positioned exactly at the sleeper supports. The two first pinned–pinned frequencies are situated approximately at 950 and 2200 Hz (UIC60-rail and sleeper spacing of 0.60 m). To attenuate these vibrations, the Department of MEMC at the VUB has developed a dynamic vibration absorber called the Double Tuned Rail Damper (DTRD). The DTRD is mounted between two sleepers on the rail and is powered by the motion of the rail. The DTRD consists of two major parts: a steel plate which is connected to the rail with an interface of an elastic layer, and a rubber mass. The two first resonance frequencies of the steel plate coincide with the targeted pinned–pinned frequencies of the rail. The rubber mass acts as a motion controller and energy absorber. Measurements at a test track of the French railway company (SNCF) have shown considerable attenuation of the envisaged pinned–pinned frequencies. The attenuation rate surpasses 5 dB/m at certain frequency bands.

1. Introduction

Railway (tram, train and subway) induced vibrations are a growing matter of environmental concern. The rapid development of transportation, the increase of vehicle speeds and vehicle

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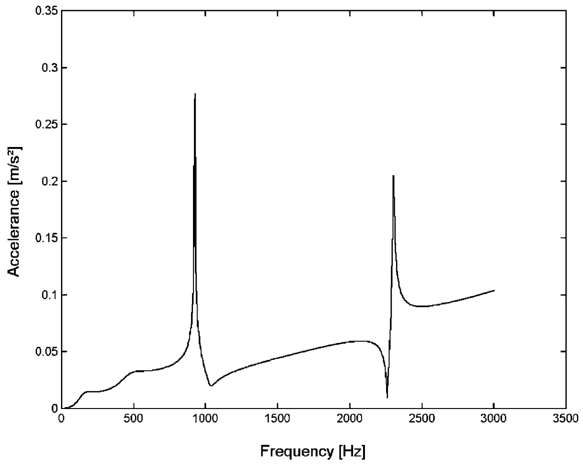

Fig. 1. Accelerance due to a harmonic excitation of a normal track layout (Trade, KUL).

weights have resulted in higher vibration levels. In the meantime vibrations that seem to have been tolerated in the past, are now more often considered to be a nuisance. Numerous solutions have been proposed to remedy these problems. Railpads are a standard part of the track, sub-ballast mats are a growing feature and under-sleeper mats are in use on ballastless track. However, all of these measures however only influence the dynamic behaviour of the rail when the rail transmits vibrations through the sleeper. Fig. 1 shows the accelerance due to a harmonic excitation of the rail in the 0–3000 Hz frequency range. The graph is the result of a simulation with an analytical track model. The Trade software has been developed by the Catholic University of Leuven, Belgium. The position of the excitation and the calculated accelerance is in the middle of a sleeper spacing. The calculation is based on a traditional railway track consisting of a UIC60 rail, concrete sleepers with a sleeper spacing of 0.60 m and a ballast foundation. Numerous resonance frequencies can be seen. At approximately 950 Hz (the actual frequency depends on the rail type, the fixation of the rail and the supporting structure) the first pinned–pinned vibration eigenmode occurs. At this resonance or eigenfrequency of the railway track, the nodes of the rail's mode shape coincide with the positions of the sleepers. There is zero amplitude of the rail at the sleepers and thus all of the measures to isolate or dampen the vibrations, as described above, are not activated. A second pinned–pinned frequency can be found at approximately 2200 Hz. At this frequency the same phenomenon occurs for a higher harmonic. [1,2].

The subject of this paper is aimed at these two eigenfrequencies (see Fig. 2) whereby no vibrations are transmitted through the railway track in the ground, although the rail itself still resonates at these pinned–pinned frequencies. The important displacements of the rail at these eigenfrequencies result in considerable levels of noise emissions. Fig. 3 shows, over the whole frequency range, the contribution of the wheels, the rail and the sleepers. It is an example of the calculations by Thompson with the TWINS software [3,4]. At the pinned–pinned frequencies the rail is not only the major contributor to the total level of emitted noise, but the noise levels from the rail are the highest throughout the frequency range. Therefore it is important to concentrate on the pinned–pinned frequencies and to develop means to reduce the effects of these mechanisms.

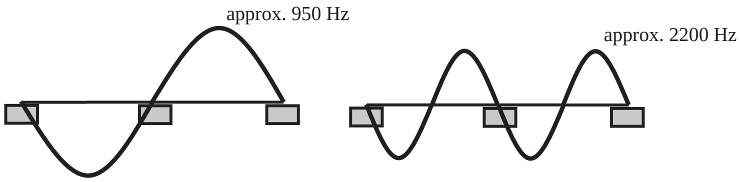

Fig. 2. Mode shapes at pinned–pinned frequencies.

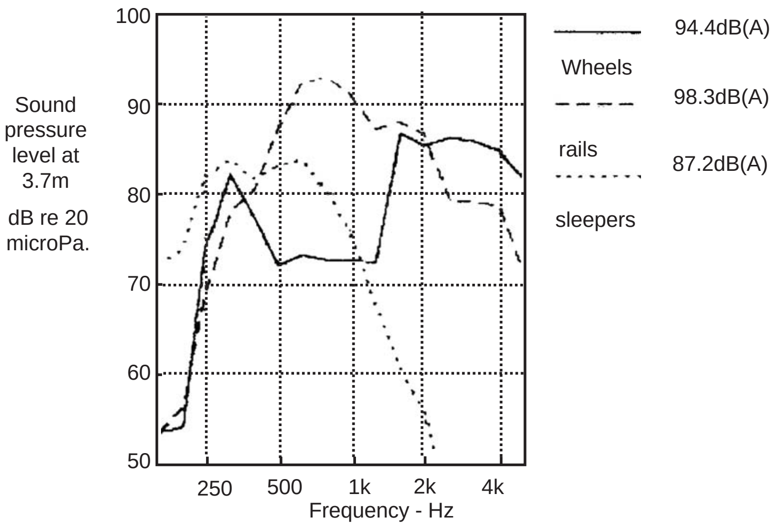

Fig. 3. Noise emissions of the track [3,4].

To attenuate the pinned–pinned frequencies, the Department of Mechanics of Materials and Constructions of the Free University of Brussels has developed a dynamic vibration absorber (DVA) called the Double Tuned Rail Damper (DTRD). The development was achieved in close co-operation with CDM, Overijse, Belgium. The DTRD is mounted on the rail between two sleepers and is powered by the motion of the rail. This paper describes the tests performed on a DTRD. The development is concentrated mainly on the attenuation of the vertical vibrations. Results however show that the DTRD is also effective for the attenuation of the lateral vibrations.

2. Description

2.1. Working principle of a DVA

Applying a force that counteracts the movement of the object can reduce the amplitude of a vibrating object [5,6]. In this case the object is the rail. In Fig. 4, the rail is vibrating at the first pinned–pinned frequency. When it is freely vibrating, the amplitudes can become important. This results in the emission of high noise levels. To reduce the amplitudes, a force is necessary. The movement of the rail itself generates this force.

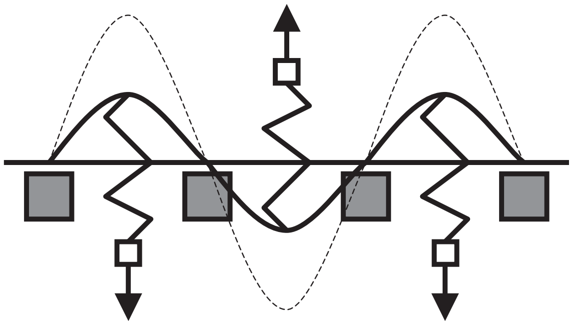

Fig. 4. Working principle of a dynamic damper.

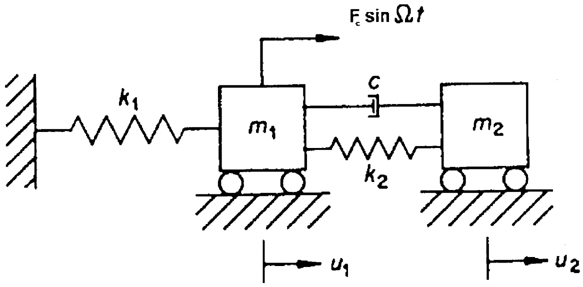

Fig. 5. Undamped construction (modelled as a 1 d.o.f. system) and an attached (1 d.o.f.) dynamic vibration absorber.

The movement of the rail puts a mass in motion that is connected to the rail by a spring. This spring–mass system is called a dynamic vibration absorber (DVA). To work properly, the mass and the stiffness of the spring have to be designed so that the eigenfrequency of the DVA is the same as the frequency of the vibrating object. For optimal efficiency the DVA has to be placed where the movement of the rail is maximal. This construction can be modelled as an equivalent system with one degree of freedom (d.o.f.) as seen in Fig. 5.

The mass m_2 of the added DVA is small compared to the mass of the construction m_1. The efficiency of the DVA can be demonstrated by plotting the dynamic amplitude versus the frequency. The efficiency is strongly dependent on the modal damping ratio value of the tuned DVA. Fig. 6 shows some examples of dynamic amplitude curves of the construction divided by the static amplitude for different damping ratios of the added vibration damper (mass ratio construction/damper = 0.05). The physical principle of a tuned vibration absorber is that the vibration mode of the 1 d.o.f. construction model is divided into two vibration modes: one mode with the amplitude of the construction in phase with the absorber amplitude, and another mode with the absorber amplitude in anti-phase.

2.2. Details of the DTRD

The DTRD, as depicted in Fig. 7, consists of three major parts: a steel plate (1), connected to the rail with a visco-elastic interface (2), and two rubber blocks (3). The rubber blocks consist of

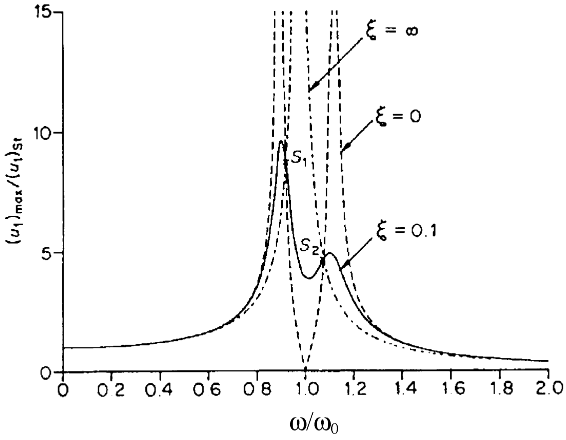

Fig. 6. Efficiency of a tuned vibration absorber for different damping ration values (u_1 is the vibration amplitude of the construction).

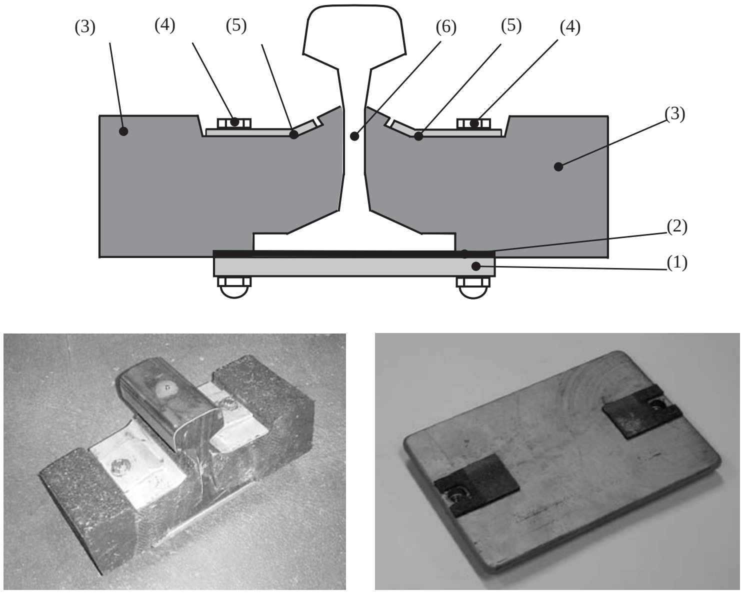

Fig. 7. Detailing of the DTRD.

recycled automobile tyres that are grained and moulded with a PUR-matrix. Two bolts (4) and metal plates (5) are used to push the rubber blocks against the rail (6). The rail is wedged between the visco-elastic layer and the rubber blocks.

The characteristics of the DTRD are:

- The DTRD has a total mass of 9 kg. Since the sleeper spacing is assumed at 60 cm, the mass per meter rail length is 15 kg.

- The DTRD is designed to be fitted on the rail without any modifications needed to the rail, to the sleepers, nor to the supporting ballast layer.
- The torque of the bolts is set to 4 kg m for optimal working conditions.
- The electrical isolation is maintained.
- The radiation of sound by the DTRD itself is negligible.

2.3. Working principle of the DTRD

An extrapolation of the 1 d.o.f. system to a 2 d.o.f. system is necessary when the two pinned–pinned frequencies both need to be damped. The same principles (see Fig. 8) can be applied to an undamped construction, modelled as a 2 d.o.f. system using a 2 d.o.f. tuned DVA attached to the construction; hence the name 'Double' tuned rail damper.

The necessary reaction force to reduce the amplitude of the vibrating rail at the two pinned–pinned frequencies is mainly generated by the steel plate. The bending mode shapes are the eigenmodes that give the highest reaction force. The vertical rigid body of the steel plate mode would have given an even higher reaction force, but can not occur since the bolts prohibit this vibration mode. Therefore, the steel plate is designed to have two specific eigenfrequencies, with bending mode shapes that coincide with the pinned–pinned frequencies of the rail. The rubber blocks act both as motion controllers and energy absorbers. The vibration of the blocks partially dissipates the energy that is transferred into it from the rail.

The first resonant frequency of the bottom plate separately is tuned to a frequency of 890 Hz and is associated with a bending mode shape. The second resonant frequency is tuned to a frequency of 2090 Hz and is associated with a transverse bending mode shape. Fig. 9 shows both mode shapes.

2.4. Placement of the DTRD

In order to have maximum effect at the first and the second pinned–pinned frequency, the DTRD has to be placed between the sleepers, at a position which is 3/8 of the sleeper spacing

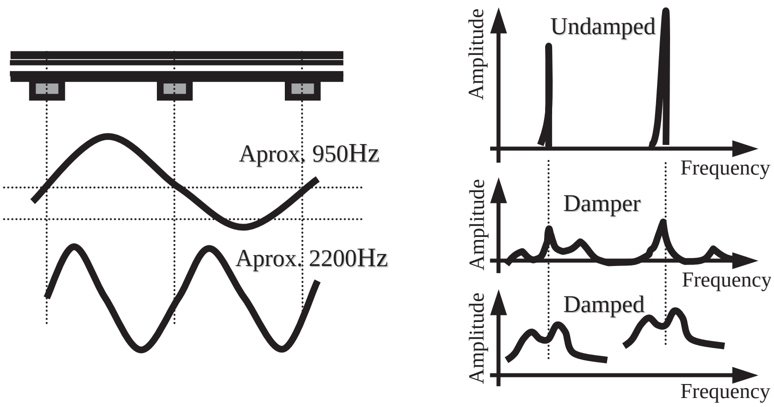

Fig. 8. Working principle of the DTRD.

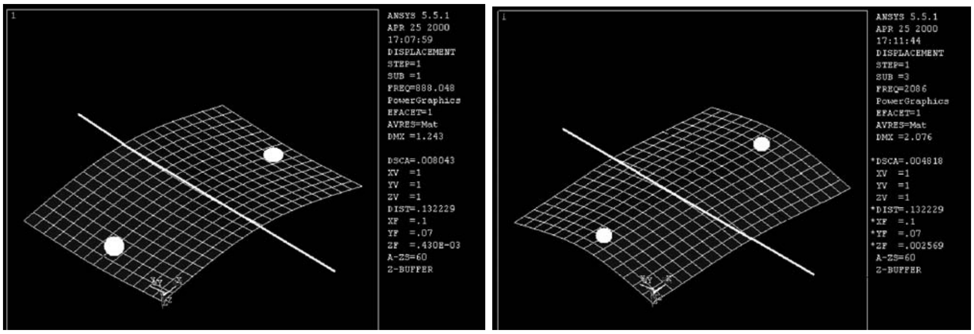

Fig. 9. Mode shapes of the bottom plate.

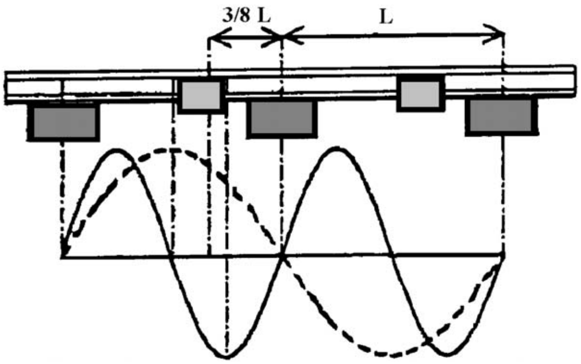

Fig. 10. Positioning of the DTRD on the rail.

(i.e., 0.225 m) from the sleeper. At this distance, the vertical displacements are close to their maximum value for both pinned–pinned frequencies as seen in Fig. 10. Thus the damper has maximum efficiency at both pinned–pinned frequencies considered.

3. Experimental study

3.1. The dynamic vibration test

The dynamic vibration test is used to determine the dynamic characteristics of the damper. The DTRD is mounted on a rail with the same length as the damper. The rail and damper are hung with strings to fixed connection points above. The boundary conditions influence the results of the test significantly. Therefore multiple fixations of the strings on the DTRD have been tested to find the optimal set-up as seen in Fig. 11.

A shaker with a signal generator is connected to the top of the rail by means of a stinger. A periodic chirp is chosen. This is a periodic repeating signal, which combines all frequencies, assembled in bandwidths, within the targeted frequency range of 300–2500 Hz. A sensor is placed between the stinger and the rail to measure the force on the rail.

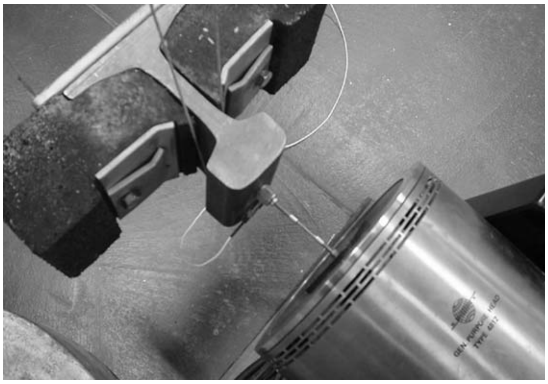

Fig. 11. Dynamic vibration test set-up.

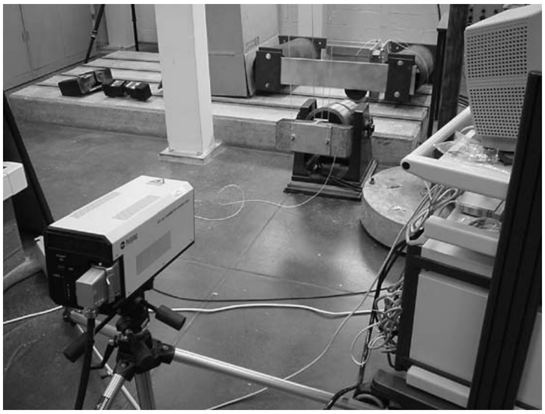

Fig. 12. Set-up of laser Doppler vibrometer.

The Polytec (PSV-300) Laser Doppler Vibrometer (LDV) is placed at a certain distance from the DTRD (Fig. 12). A built-in camera gives an image on a monitor of the DTRD. On this image, the measurement points of the laser are defined.

The LDV works by laser interferometry. A diffractor splits the laser beam. Part of the beam stays inside the machine, while the other part is reflected onto the DTRD. The two beams are then reassembled. The interference between the two beams is a measure for the speed of the surface on which the second part of the beam is reflected. The laser scans, one by one, the points defined earlier. The measurements are transmitted to a computer to calculate the frequency response functions (FRF).

The resulting FRF is shown in the Fig. 13. The eigenfrequencies can be clearly distinguished in the frequency interval of interest (750–2250 Hz) at 990 and 2304 Hz. The software also allows the visualization and even the animation of the corresponding eigenmodes as seen in Figs. 14 and 15.

At 990 Hz, the first bending mode appears in the steel plate (Fig. 14). The mode shape is similar to Fig. 9. At 2304 Hz, the second bending mode appears. The eigenmode, seen in Fig. 13, is the bending mode as calculated and shown in Fig. 9. The measured eigenmodes correspond with

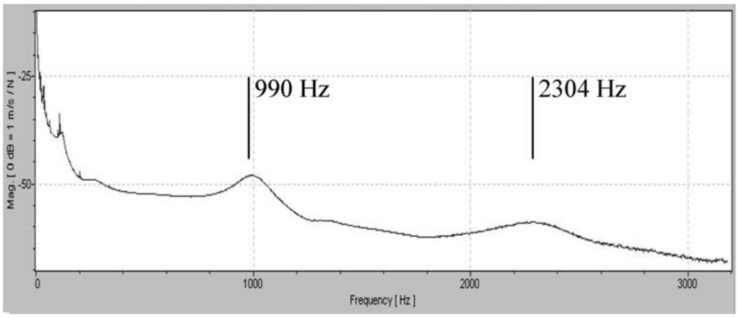

Fig. 13. The frequency response function.

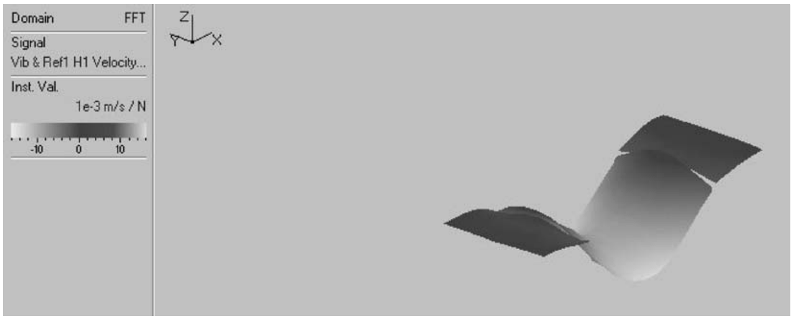

Fig. 14. Eigenmode at 990 Hz.

Fig. 15. Eigenmode at 2304 Hz.

the expected and calculated modes. The numerical difference between FE calculations and the measurements is due to suppositions made in order to simplify the FE model.

3.2. The impact test

The impact test is aimed specifically at the pinned–pinned frequencies. The performance of the DVA is checked by comparing the damping ratios of undamped and dampened rail

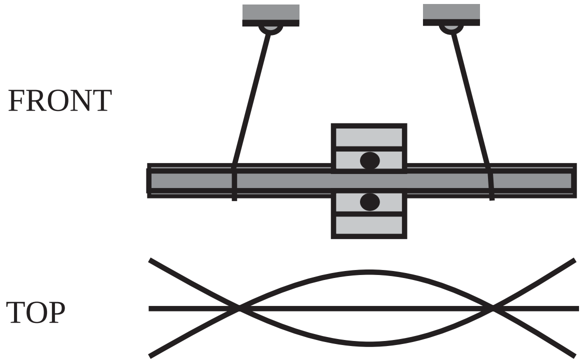

Fig. 16. Schematic test set-up impact test.

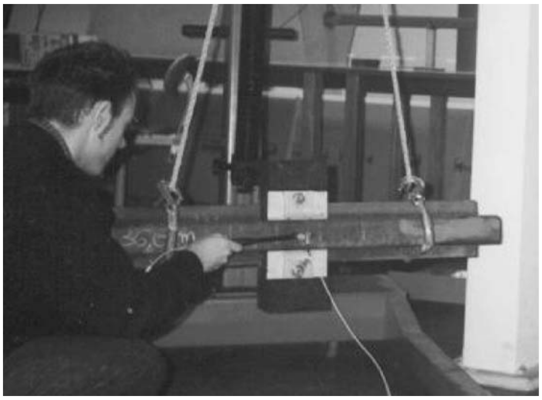

Fig. 17. Test set-up impact test.

lengths. The set-up of the test consists of a rail, hung sideways with elastic cord to a fixed structure. This configuration simulates free–free boundary conditions. The rail is cut to a predetermined length with a first bending eigenfrequency in the vertical plane of the rail coinciding with one of the two pinned–pinned frequencies of interest. A long rail section of 0.96 m and a short section of 0.66 m in length are cut. These rail lengths correspond respectively to the first and the second pinned–pinned frequency. The connections between the rail and the cables, with which the rail is supported, are made at the nodes of the first eigenmode of the rail, so no interference between the environment and the vibration of the rail occurs (Fig. 16).

One accelerometer (PCB) is placed in the middle on the top of the rail, and is connected to a workstation. The rail is impacted at the middle by a hammer to have maximum effect as can be seen in Fig. 17. The force of the impact is also inputted to the workstation. The LMS CadaX software visualizes the FRFs and calculates the damping values at the frequencies defined by the operator. Two reference tests, one for each rail length, consisted of a test set-up without a DTRD attached.

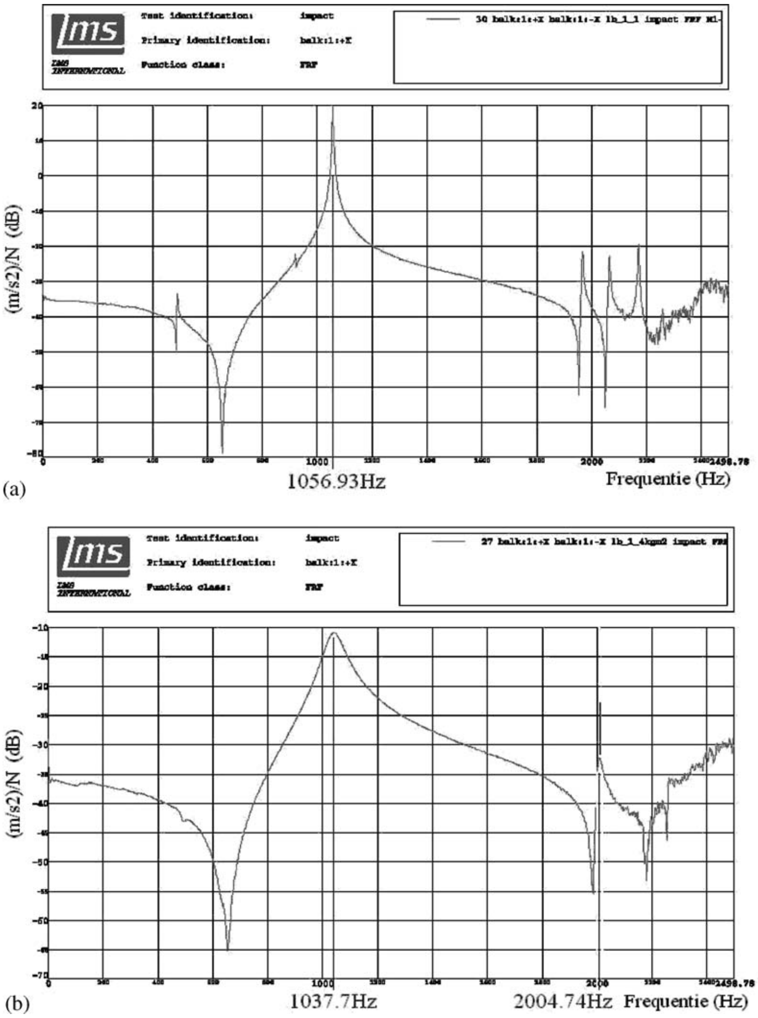

Fig. 18. First pinned–pinned frequency without damper (a) and with damper (b).

The FRFs without a DTRD are shown in Figs. 18 and 19. The DTRD is fixed to the rail for the second part of the experiment. The DTRD is placed in the middle of the rail length. There, the diplacements are maximal.

The results, represented in the graphs, show an decrease of 30 dB and even 40 dB in vibration levels at the first and the second pinned–pinned frequency respectively. These results are however just an indication of the capabilities of the dynamic damper. In reality even the pinned–pinned frequencies are already damped. The connection between the rail, the railpad and the sleeper is the source of the damping, even at the pinned–pinned frequencies, so the results in-situ will be less spectacular.

Table 1 gives an overview of the damping values at the target frequencies. Without the DTRD, the rail is virtually undampened. With the DTRD the damping attains 3.46% and 6.25% at the first and the second pinned–pinned frequency.

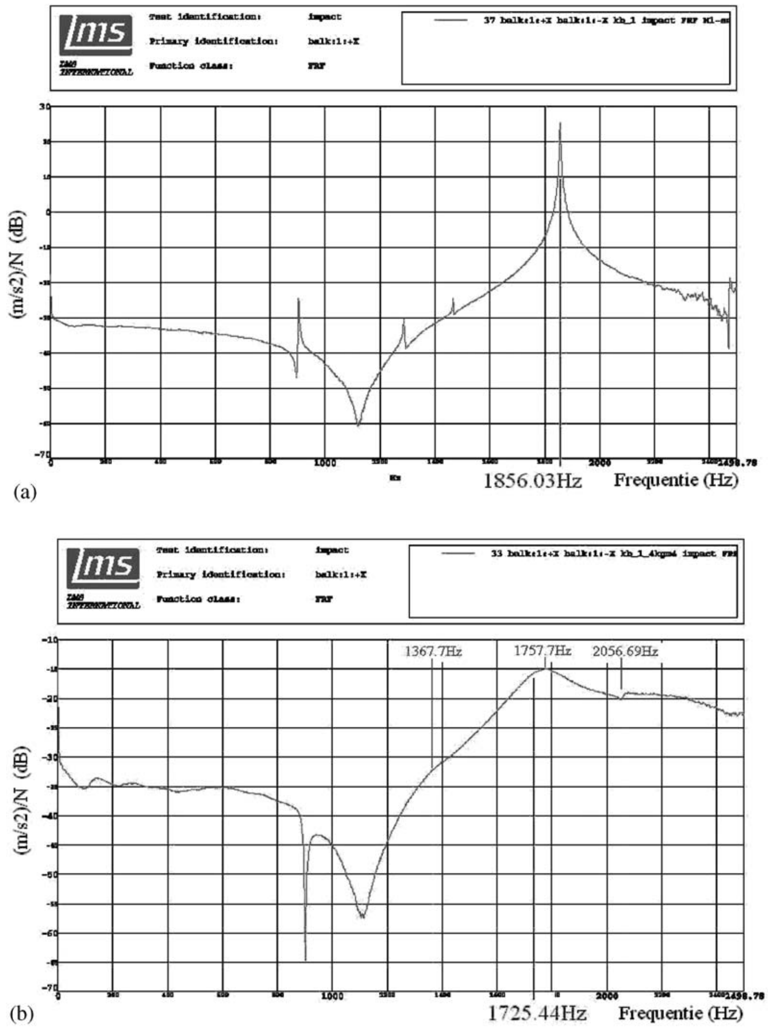

Fig. 19. Second pinned–pinned frequency without damper (a) and with damper (b).

Table 1
Summary results of the impact test

DTRD	First pinned–pinned (long rail)		Second pinned–pinned (short rail)	
	Eigenfrequency (Hz)	Damping (%)	Eigenfrequency (Hz)	Damping (%)
No	1057	0.02	1856	0.03
Yes	1038	3.46	1758	6.25

3.3. Short ballasted rail track test set-up

A ballasted test track with two 1.5 m long rails was set up with the following characteristics (Fig. 20):

- rail type UIC-60, supported on three concrete sleepers;
- sleeper type concrete bi-bloc VAX U41, with an inter-sleeper distance of 0.60 m;
- ballast type porphyry 2.5–5 under the sleeper with a thickness of 0.35 m;
- rail pads type CDM 66009 with a NABLA fixation system.

The test track transfer functions are measured on a non-equipped track and on a track equipped with DTRD. The rail was divided into a regular grid of 11 points 0.15 m apart. Four accelerometers (two horizontal and two vertical) were placed on the track:

- one horizontal and one vertical half-way between two sleepers;
- one horizontal and one vertical at 1/4 of the way (0.15 m) between two sleepers.

The accelerometers and an impact hammer with a load cell are connected to a CadaX data acquisition system from LMS. 44 transfer functions are measured for each configuration (4 fixed position accelerometers $\times$ 11 hammer impact positions). Fig. 21 shows the four accelerometer

Fig. 20. The short ballasted rail track test set-up.

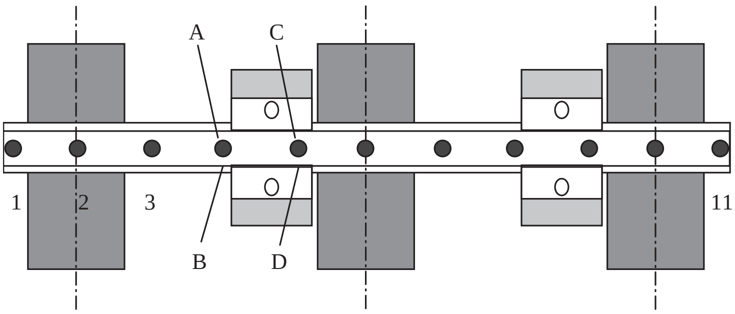

Fig. 21. Accelerometer positions (with an attached DTRD); 1–11: hammer impact positions; A–D: accelerometer positions.

Table 2
Summary results on the short ballasted rail track test

Mode shape	Frequency (Hz)	Damping ratio (%)	Frequency (Hz)	Damping ratio (%)	Damping increase (%)
	Without DTRD		With DTRD		
First vertical pinned–pinned frequency	985	3.98	976	7.30	3.30
Second vertical pinned–pinned frequency	2610	1.03	2650	5.70	4.70
First horizontal pinned–pinned frequency	575	1.03	570	2.50	1.50
Second vertical pinned–pinned frequency	953	0.73	954	1.30	0.60

positions and the 11 hammer impact positions. Table 2 gives the measured frequencies and damping ratios.

Due to the short length of the test track, many wave reflections occur in the rail. Furthermore, the vibration nodes of the vertical bending mode shapes are not on the central axis of the sleepers (to obtain the undamped pinned–pinned frequencies). As a consequence, the measured vertical bending mode shapes are mainly damped out by the fixing system at the sleeper axis. There are also some discrepancies between the pinned–pinned frequencies aimed for and the measured frequencies, for example 2610 Hz at the second pinned–pinned frequency. The next phase in the development of this short ballasted rail test track is thus to minimize the wave reflections and to place the vibration nodes at the central axis of the sleepers by changing the boundary conditions of the rail ends. Nevertheless the increase of damping is very noticeable.

3.4. The in situ test (SNCF)

The in situ tests were carried out at French Railway (SNCF) test centre at Saint-Ouen, France. There, the SNCF has the necessary equipment to test every component of a railway track. The DTRD was installed on their test track to evaluate the in situ dynamic performances in view of SNCF's SOFEB project (Optimized Solutions for Low Emission of Noise).

The characteristics of the SNCF test track are:

- UIC 60 rail;
- NABLA fixations;
- bi-bloc sleepers;
- 0.60 m sleeper spacing;
- shore 70 railpads with a thickness of 7 mm; and
- 0.36 m ballast thickness.

Two accelerometers were attached to the rail, one at a sleeper, the other between two sleepers. The rail was excited by an impact hammer. The impacts were made at increasing distances from the accelerometers. A reference measurement, without the installation of the DTRDs, was the basis for comparison. After the reference measurement, the DTRDs were placed following the installation prescription as seen in Fig. 22.

Fig. 22. SNCF test set-up at Saint-Ouen.

The SNCF used the wave decay rate to evaluate the dynamic performance of the DTRD. The wave decay is the decrease in value of the amplitude of the acceleration in dB:

$$\Delta = 20\log_{10}\left(\frac{a(x_1)}{a(x_2)}\right) = 20\log(a(x_1)) - 20\log(a(x_1)). \tag{1}$$

The wave decay rate is the ratio of the wave decay to the distance between two points of measurement:

$$\frac{\Delta}{\Delta x} = \frac{20\log_{10}\left(a(x_1)/a(x_2)\right)}{(x_1 - x_2)} = \frac{20\log(a(x_1)) - 20\log(a(x_2))}{(x_1 - x_2)}. \tag{2}$$

The wave decay rates were calculated for third octave bands by means of a linear interpolation.

After the installation of the DTRD, the results were much improved, not only in the vertical direction (see Fig. 23) but also in the lateral direction (see Fig. 24). In vertical direction the minimal WDR increased from 0.9 to 2.4 dB/m; in lateral direction the WDR increased from 0.1 to 0.6 dB/m. The continuous line indicates the requirements by SNCF. Although a significant improvement of the results is attained in comparison with the reference, the results are below the SNCF requirements.

4. Conclusions

The DTRD was specifically designed to reduce the vibration levels at the pinned–pinned frequencies. In order to do that, a force counteracting the movement of the rail, was applied. This force was generated by a DVA (the 'Double Tuned Rail Damper'). The eigenfrequencies of the DTRD coincided with the two pinned–pinned frequencies considered. Furthermore, the

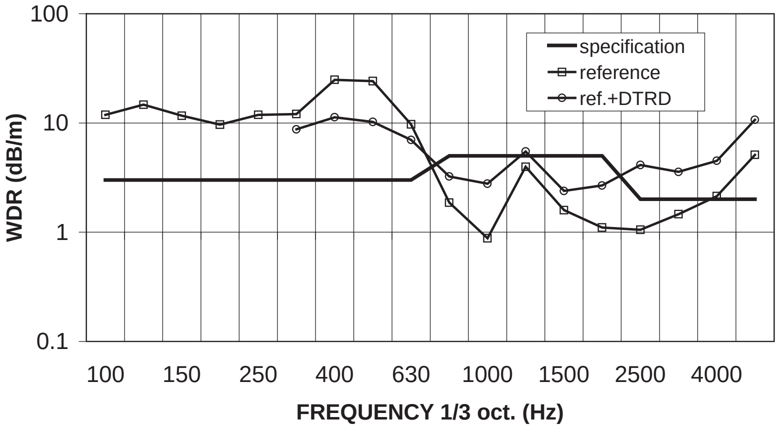

Fig. 23. Vertical wave decay rate (SNCF).

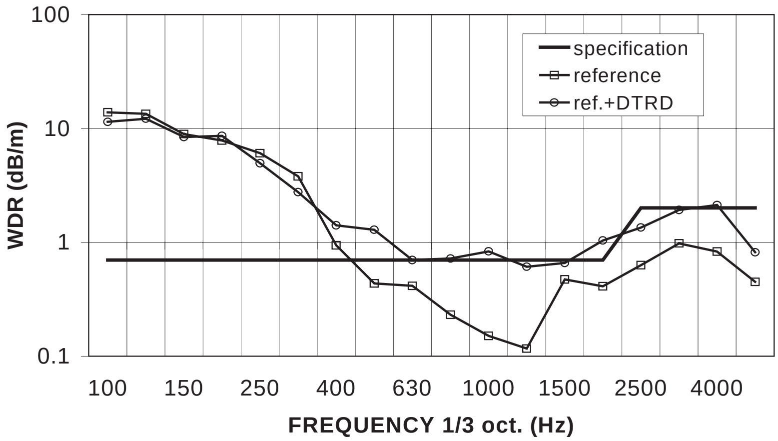

Fig. 24. Lateral wave decay rate (SNCF).

corresponding eigenmodes of the DTRD were thus chosen so that the vertical reaction force of the damper was maximal.

Measurements at a test track of the SNCF have shown considerable attenuation of the envisaged pinned–pinned frequencies. There are however some difficulties to overcome. In situ testing revealed that although a significant increase of the wave decay rate has been recorded, the specification demanded by the SNCF has not yet been met. The wave decay rate in the 500–2500 Hz frequency range has to be improved.

The first way to accomplish this is to measure the exact pinned–pinned frequencies of the track on which the damper will be applied. The eigenfrequencies of the damper can then be tuned more precisely to the requirements. The second way is by optimizing the vibration modes of the DTRD in order to amplify the reaction force on the rail. With these measures the required wave decay rates, and possibly better, can be attained. Therefore the research on the DTRD will be focussed on the physical interpretation of the interaction between the different components of the DTRD. Numerical modelling can be an additional tool for this.

Acknowledgements

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