TESTING THE DYNAMIC PROPERTIES OF RESILIENT TRACK COMPONENTS AT FREQUENCIES CRITICAL TO NOISE AND VIBRATION PERFORMANCE

Shimada, S. (1), Anderson, D. C. (2)

(1) Arup Acoustics, Sydney (formerly Rail Infrastructure Corporation)
(2) Rail Infrastructure Corporation, Sydney

Abstract

Resilient track components play an important role in operational railway vibration control. They are often used for controlling railway noise and vibration at frequencies from 5 Hz to 500 Hz.

In-track vibration control is largely determined by the dynamic stiffness and damping characteristics of resilient components. Elastomeric materials, which are used extensively in rail applications, are known to exhibit a variety of frequency-, amplitude- and preload-dependent stiffness properties.

Procurement requirements for resilient track components generally refer to laboratory-tested static stiffness and also to dynamic stiffness values at up to 20 Hz, but neglect component performance at higher frequencies. This paper reviews the potential impact of this limitation, and discusses options for future testing methodologies at higher frequencies. These include an impact test, and a combination of small-scale material testing with Finite Element Analysis.

The conclusions of the study are that:

- Current specifications, standards and testing methodologies fail to encompass dynamic performance at important frequencies above around 25 Hz.
- As a result, complying track fasteners could result in vibration performance variations of up to 10dB.
- An impact test method may provide a practical way to address this limitation. Further research is recommended to confirm and refine this method.

Nomenclature

F	Input force (N)
m	Mass (kg)
t	Time (s)
k	Stiffness (N/m)
c	Damping coefficient (N s/m)
$\zeta = \dfrac{c}{2\sqrt{km}}$	Damping ratio
x	Displacement (m)
ω	Radian frequency (rad/s)
$\omega_n = \sqrt{k/m}$	Natural undamped frequency (rad/s)

Introduction

Resilient track components are increasingly selected and installed for the specific purpose of achieving noise and vibration criteria at locations affected by railway operations. These criteria generally encompass the following issues:

- Perceptible vibration within buildings (typically in the frequency range from 5 Hz to 50 Hz).
- Groundborne (or re-radiated) noise within buildings (typically 20 Hz to 200 Hz).
- Structure-radiated noise from railway bridges and viaducts (typically up to 500 Hz).

Dynamic stiffness and damping characteristics of resilient components are fundamental to in-track vibration performance. The stiffness of an elastomeric baseplate is known to increase with increasing frequency and/or reducing temperature or dynamic amplitude. Pre-load (such as that due to the dead load of a train) can also affect stiffness characteristics.

The extent of these non-linear effects depends on the elastomeric material in question. Procurement requirements for resilient track components generally include reference to static stiffness criteria and also to laboratory-tested dynamic stiffness values for frequencies up to 25 Hz.

This paper:

- Reviews the potential impact of this limitation.
- Reviews international dynamic testing standards and methodologies in use or under development.
- Discusses options for future testing methodologies at higher frequencies.
- Presents results of Finite Element Analysis and small scale material testing.

Railway Noise and Vibration

Background

Train noise and vibration is generally categorised into three types: groundborne noise and vibration, structure-radiated noise and vibration, and airborne noise.

Groundborne noise is usually in the 30~250 Hz frequency range. Sound waves are transmitted through soil and bedrock and may be felt as perceptible vibration at large amplitudes and low frequencies up to 80 Hz, or can travel through the ground and into the air as higher frequency audible sound. Structure-radiated noise is sound waves caused by vibrating structural components, at frequencies typically ranging from 30 to 350 Hz. Airborne noise is generated at the wheel-rail interface and propagated through the air over a large frequency range, from 50 to 2000 Hz.

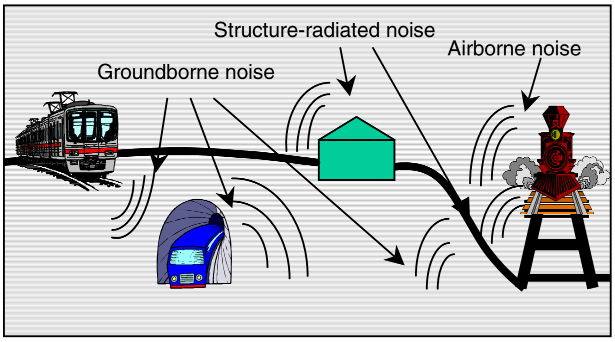

Figure 1. Categories of noise and vibration associated with railway traffic.

When no vibro-acoustic mitigation products are in use, the primary track response of typical train tracks most often occurs at 1/3 octave centre band frequencies between 31.5 and 63 Hz.

Resilient Baseplates

Resilient baseplates offer a cost-effective method to control railway noise and vibration. Vibration reduction depends on the overall stiffness of the baseplates as well as the primary response of the track without resilient baseplates.

A resilient baseplate consists of several components, set in series or in parallel. Varying amounts and particle sizes of carbon black filler in the elastomer change its hardness and its stiffness properties [1].

Figure 2. Pandrol® resilient baseplate

For example, Pandrol® VIPA baseplates comprise a studded rubber rail pad on top of a steel top plate, a second rubber pad, and a steel bottom plate in series. The rubber pads act mainly in compression. Contitech's Cologne Egg uses an elastomer component that mostly acts in shear under typical train loads.

The general aim of introducing resilient rail pads or baseplates is to reduce the primary track response. In general, this has been considered favourable in terms of environmental noise pollution.

The choice of resilient baseplate depends on its stiffness properties for these important frequencies – assuming they are known. With typical primary track responses giving vibration peaks at around 30 Hz to 80 Hz, it is apparent that the dynamic stiffness tests in this frequency range are important for ensuring resilient baseplates perform as vibration isolators in track.

Standards and Specifications

International resilient baseplate specifications generally address low frequency (up to 30 Hz) vibration in track.

In New South Wales, the Rail Infrastructure Corporation Specification C3304.2.0 (2001) [2] specifies dynamic stiffness tests conducted at around 7 to 15 Hz. German Standard E DIN 45673-1 (1998-07) [3] notes that dynamic stiffness is only measured to 20 Hz. Extrapolations up to 50 Hz assume a logarithmic relationship between stiffness and frequency.

It is well known that elastomers used in resilient baseplates exhibit non-linear elastic behaviour at high frequencies. That deviation may occur in the frequency range that is easily perceptible to humans.

The current standard of testing up to 25 Hz does not address critical frequencies in the rail traffic vibration spectrum between 30 and 80 Hz.

Existing high frequency test rigs

There have been many attempts to design high-frequency test rigs for rail-baseplate assemblies.

DD ENV 13481-6:2002 [4] defines *transfer stiffness* as the dynamic stiffness in the range of 25-400 Hz. The direct and indirect methods for measuring transfer stiffness both use a shaker operating over the range 25 to 400 Hz, and a loading frame to apply a preload of 25 to 40 kN (see Figure 3).

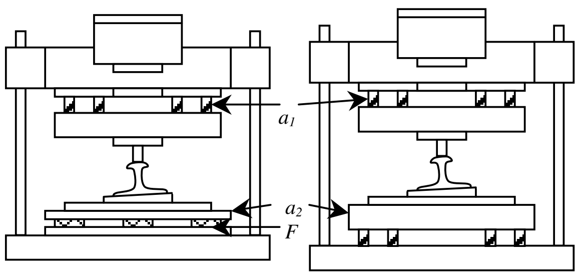

Figure 3. Transfer stiffness test rigs: direct (left) and indirect (right) methods

The Standard states that a standard deviation of about 20% can be expected from this test procedure, indicating that this type of testing is very sensitive and difficult to reproduce. The procedure for estimating the vibration velocity in the track suggests a rather haphazard approach based more on aiming at goal values rather than on calculating empirical results from known data. This test has potential, but needs a more consistent approach.

Pandrol® [5] compare dynamic stiffness data from in-track and various high frequency laboratory tests. The Pandrol® report concludes that there are significant differences in test conditions. The greatest discrepancies in results are those for low amplitude, high frequency inputs, which represent typical rail-related excitation. Discrepancies may be due to different reference values, or fastening systems, or corrections for test rig response to excitation.

Impact Test

A high frequency test rig

Current Standards and Specifications omit dynamic stiffness tests of entire resilient baseplates at frequencies over 20 Hz. However, the primary response of typical tracks occurs at frequencies between 30 and 80 Hz. An impact test may provide a cost-effective rig that yields easily repeatable, reliable results. A possible impact test rig is described below.

Proposed test rig description

The impact tester uses an instrumented hammer of known mass dropped from a known height, so that the force and acceleration of the impact is known. By definition, the impact load will result in a range of input frequencies

The materials and dimensions of the components in the test rig have been chosen to ensure that the test rig can withstand the static loads as well as avoid resonance at the range of frequencies expected from the impact.

Figure 4 shows a schematic diagram of the test rig.

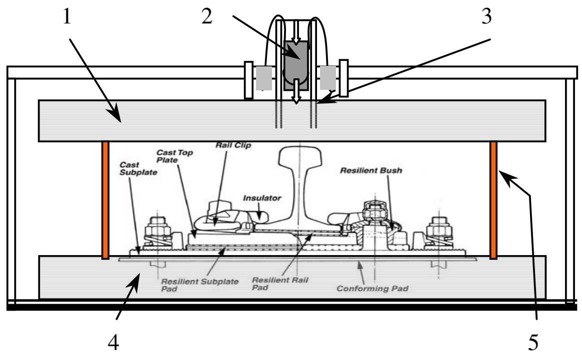

Figure 4: Impact test rig (with Pandrol baseplate)

This impact test rig comprises:

(1) Very stiff beam or top plate such as mild steel, with a resonance frequency above the frequency range of interest to the test under the applied pre-load.

(2) Known weight with a highly resilient pad at the base, dropped from preset height to produce a known impulse load.

(3) Guide rails with catch mechanism: rope with counterweights, held on frame, to ensure vertical load.

(4) Very solid / inert base to approximate the "rigid foundation" assumed in theory.

(5) 4 bolts, used to apply pre-load representative of typical train axle load, and instrumented with velocity transducers to measure motion under applied loads.

Existing impact tests

Impact test rigs are already available and are used in impact strength tests. Such a test rig and procedure is described in British Standard BS EN 13146-3:2002 [6].

In this test, an impact load is applied via a falling mass on the head of a rail fastened to a concrete sleeper. The test is performed on a reference assembly with a standard rail pad, and with the test pad, for comparison. The British Standards test is to calculate impact attenuation, but could be appropriate for finding dynamic stiffness or transfer stiffness over a wide range of frequencies, especially high frequencies.

Analysis of impact test results

Standard dynamic methodology (eg Fourier or Laplace analysis) can be used to analyse the output from the impact tests on the rail-baseplate assembly. The data can be analysed in terms of input/output force or acceleration. The input force F_{in} is known from the weight of the dropped mass. The frequency range can be discretised (eg in 1/3 octave bands) for application of Laplace transforms.

Equation of motion: $m\ddot{x} + c\dot{x} + kx = F(t)$

or $\ddot{x} + \frac{c}{m}x + \frac{k}{m}x = \ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n{}^2 x = F(t)$

For the underdamped case ($\zeta < 1$), using Laplace Transformation:

$$x(t) = \frac{e^{(i\omega_n\sqrt{\zeta^2-1})t}}{2\omega_n\sqrt{\zeta^2-1}}\left[\frac{\dot{x}_0 + (\zeta + \sqrt{\zeta^2-1})\omega_n x_0}{e^{\zeta t}} - \frac{\dot{x}_0 + (\zeta - \sqrt{\zeta^2-1})\omega_n x_0}{e^{-\zeta t}}\right]$$

Using the Euler identity $e^{i\omega t} = \cos\omega t + i\sin\omega t$, and re-writing in terms of m, k and c, the equation of motion can be rewritten as:

$$x(t) = e^{-\sqrt{\frac{k}{m}}t}\left[\frac{2m\dot{x}_0 + cx_0}{\sqrt{c^2-4mk}}\sin\left(\sqrt{c^2-4mk}\right)\cdot t + x_0\cos\left(\sqrt{c^2-4mk}\right)\cdot t\right]$$

In this case the terms m, k and c refer to equivalent mass, stiffness and damping coefficients of the entire assembly. System-specific values for m, k, and c are appropriate for this application, since the material values depend on input frequency and amplitude as well as materials.

It must be stressed that these models are based on single frequency inputs so the stiffness and damping values can be considered constant in each frequency case. These values may be compiled in a table for each amplitude and frequency of loading, and applied for each dynamic load case. The same property table can be used in the finite element analysis or lumped mass approach.

Small-Scale Tests with FEA

Alternative test and analysis

Inputs from trains at high frequencies are due to small scale irregularities on the rail and wheel interface, and have small amplitudes. These conditions make testing a large sample such as the rail-baseplate assembly difficult.

In the absence of a full-scale test rig, Finite Element Analysis may be used in conjunction with small-scale material tests.

Methodology

A Dynamical Mechanical Analyser (DMA) uses two small samples sandwiched between movable 10mm×10mm steel plates. DMA is suited to small amplitudes and frequencies up to 250 Hz, in both shear and compression.

For this study, DMA was used to test small samples from a Pandrol resilient baseplate top railpad. Excitation frequencies and amplitudes, as well as temperature, were varied, to measure the dynamic stiffness and damping properties of the elastomer under varying dynamic loading conditions. These stiffness and damping properties were used in a Finite Element model of the baseplate assembly, to find static and dynamic stiffness properties of the entire baseplate assembly.

Material properties of elastomers

Dynamic stiffness and damping properties were measured using DMA testing. Some material property terminology used for this study needs definition.

Dynamic stress and strain are dependent on input frequency (ω), stress-strain amplitude (ε_0 and σ_0), and the time lag factor δ, which indicates the energy dissipated during a single stress-strain cycle.

The Storage Modulus, *E'*, and the Loss Modulus, *E"*, relate to the hysteresis effect, one of the consequences of the phase difference between stress and strain [7]. They are defined as:

$$E' = \frac{\sigma_0 \cos\delta}{\varepsilon_0} \quad \text{and} \quad E'' = \frac{\sigma_0 \sin\delta}{\varepsilon_0}$$

$$\tan\delta = \frac{E''}{E'} = \frac{\sigma_{viscous}}{\sigma_{elastic}}$$

The $\tan\delta$ equation represents an important property commonly used in rubber materials theory. "Tan delta," or the loss factor, is the ratio of the viscosity to the elasticity of the rubber when excited at a known frequency.

DMA test results: Frequency effects

The DMA results show that increasing the excitation frequency increases the viscous and elastic stiffness properties of the rubber. The result is that transmissibility increases at high frequencies. This dynamic stiffening is dependent on the phase angle between storage and loss moduli, which is influenced by the type and amount of filler present in the rubber.

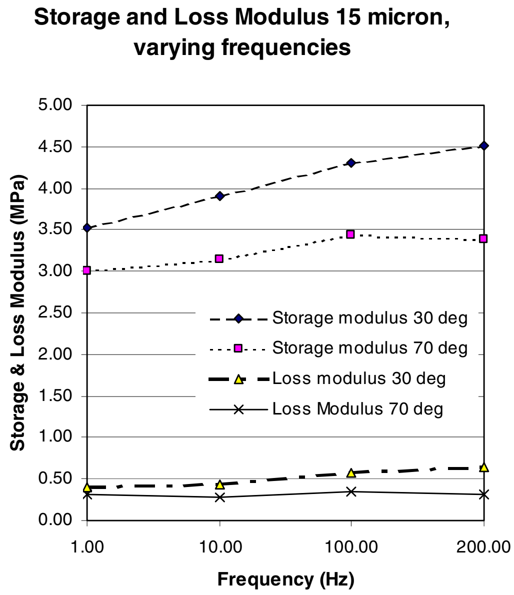

Figure 5: Shear Storage and Loss Modulus vs. Frequency plot

DMA test results: Amplitude effects

The elastic stiffness properties generally decrease as amplitude increases, as expected. The tan delta values increase with amplitude, indicating that the viscosity increases with amplitude.

Combining the effects of increasing frequency (leading to increasing elastic and viscosity) and decreasing amplitudes (increasing elastic and effective stiffness and decreasing viscosity) means that a general increase in total stiffness is expected for the higher frequency loading conditions on the resilient baseplate rail pads.

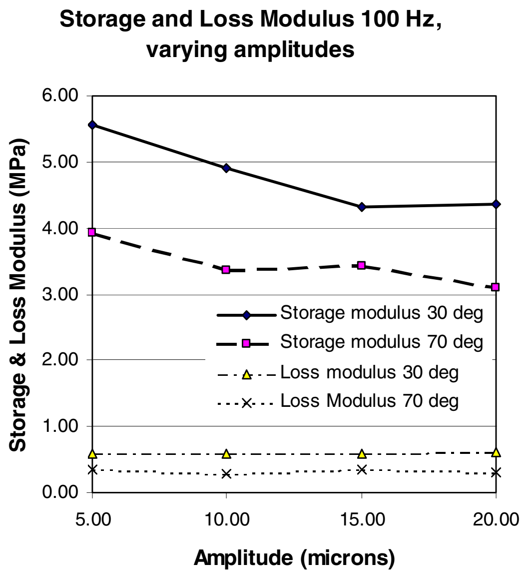

Figure 6: Shear Storage and Loss Modulus vs. Amplitude plot

Limitations of the DMA test

The DMA tester cannot apply the same amplitudes seen in the train-track environment at the higher end of the frequency range. At high frequencies, high (0.2 mm) amplitude excitation causes the rubber sample to slip between the test rig plates. Since the rubber has stiffness and damping properties that are dependent on amplitude of excitation, DMA results may not represent the dynamic properties resulting from train load conditions. However, the tan delta value found from these data is in close agreement with the values expected from extensive theoretical and empirically-based predictions.

Also, the DMA test rig cannot apply the preloads expected from real train loads. Therefore the effect of preloads on stiffness properties cannot be found via DMA, and may influence the accuracy of FEA results.

Finite Element Model assumptions and methodology

Finite element modelling allows more complex loading and stiffness models than lumped mass models. Finite element analysis has gained considerable popularity as computing power has improved. Its popularity poses the danger that incorrect modelling and subsequent incorrect results are believed without reference to simplified calculations and/or reliable empirical tests.

Strand7 finite element software was used to create a three dimensional model of the Pandrol® rail-baseplate assembly. Strand7 allows material inputs for rubber-like materials based on different rheological models.

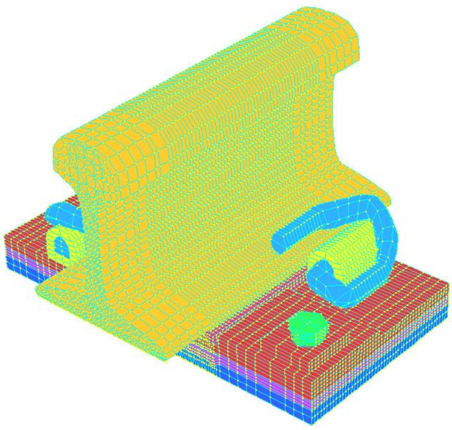

Figure 7: Finite Element model of rail and baseplate assembly

The rubber studs on the rail pads are circular in section, but were simplified in this model as hexagons, which can be tessellated and integrated with the conjoining steel parts.

It is assumed that there is no slip between the rubber and steel components. This is reasonable, since the subplate rail pad is glued to the top and bottom plate; the top rail pad is shaped to prevent slip along the lateral axis of the rail; and the rail clip holds the rail firmly onto the top rail pad and baseplate.

Similarly, it is assumed that no slip occurs between the rail clip and the clip hole, or the rail foot. This no-slip condition is the desirable condition in real applications. Any deformation should in practice be due to frictional and elastic-surface stresses, and the two surfaces should in the most part move together in creep. Longitudinal slip between the clip and the rail is not likely to be caused by regular train traffic. Therefore the no-slip condition at the clip-holes is reasonable.

The bolts are to be fully fixed, sharing nodes and edges, to the bolt holes in the sub plate only. The bolts are disconnected from the rubber pads and upper steel plate. In the real baseplate, the bolts provide lateral constraints but the top plate and rubber pads can move vertically, relative to the bolt.

The load is applied to the top of the rail in two components: a horizontal component H and a vertical component V, related by $H = 0.6V$. This is the standard condition given in testing specifications. The preload is fixed at 15 kN, 40 kN or 50 kN, to match the laboratory tests against which the FEA was compared.

The bottom surface of the steel subplate is fixed. This is the part that would be attached to the (concrete) sleeper. If the sleeper is assumed to be a perfectly rigid solid, then the force output from the bottom of the baseplate should reflect the vibration output expected at the top of the sleeper when in the track.

The rubber material properties are assigned according to classic neo-Hookean models. This model is suitable for small-amplitude excitation, as seen in medium to high frequency wheel-rail interaction.

Because of the complexity of the model, the solution time is long. This is exacerbated by the fact that non-linear properties are involved. Therefore it is simpler to input properties for the rubber that are suited to the discrete input excitation amplitude and frequency, using the materials properties tables from the DMA tests. This method allows a quasi-linear model to be used, thereby reducing the solution time.

Finite Element Analysis static stiffness results

The static loading on this model was changed to examine whether the change in static stiffness observed in the laboratory could also be seen in the finite element model. A static load of 10 kN, 20kN then 40 kN was tested in the FE model (see Figure 8).

The FE model gives results between static test results for a 66 Shore A and 48 Shore A baseplate with a 10 kN and 20 kN preload. With a 40 kN preload, the FE result is higher than laboratory test results. This is likely to be due the lack of comparable preloads in the DMA tests.

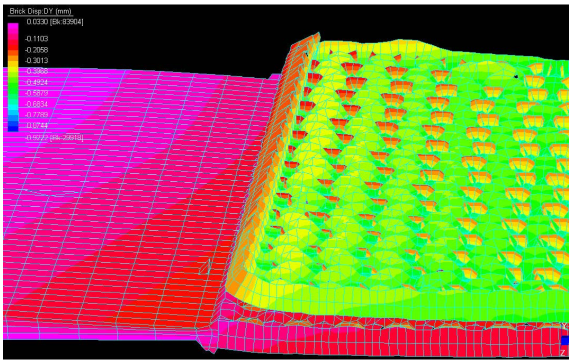

Figure 8 Detail of rubber rail pad deflections under 10 kN static load

FEA high frequency dynamic stiffness results

For this simulation, dynamic loads from 0.4 kN to 4.0 kN can be applied at 50 Hz, 100 Hz and 200 Hz.

A medium to high frequency dynamic solve in the Strand7 model has the potential to show whether linear extrapolation or a transfer stiffness model is more appropriate. However, the model still needs to be refined to account for the changing stiffness and damping properties of rubber under different loading conditions. The material properties to be entered in the model must be based on excitation amplitudes and frequencies similar to those used in current tests. Only then can the model be used as a check for various testing or prediction methods.

Finite Element Analysis validation

No computer simulation result should be accepted without some cross-reference to expected results. To verify the accuracy of the finite element model, several checks are appropriate.

Manufacturers already must perform simple and reliable static stiffness tests of the rail-baseplate assembly. The first FE model check should therefore be a simple static stiffness solution, using the same loads as defined in the tests.

The next validation check is the low-frequency dynamic simulation. Using the same preloads, loads and frequencies in the FE model as laboratory tests provides a good comparative technique to verify the FE model.

The high frequency dynamic response found using the Finite Element Model is more difficult to validate. Estimating the input from real trains is far too complicated to attempt to simulate in-track conditions. The high frequency tests described in previous pages are themselves in some doubt, as they have been found to provide inconsistent results. Comparison with lumped mass models may provide some simple model checks.

The finite element analysis also may prove helpful in determining which high frequency test rigs currently available in industry give consistent results with the finite element methodology.

Conclusion

Most current standards for resilient baseplate require dynamic stiffness testing at frequencies up to 25 Hz. High frequency test rigs currently in use or development yield inconsistent results. A reliable reference is needed to compare and assess data from different test types.

An impact test rig is of particular interest because it is relatively inexpensive to build, and it covers the range of input frequencies found in the rail environment.

The validity of finite element analysis and lumped mass model analysis methods cannot be verified without comparison with a real controlled laboratory test. The additional components, preload, and vertical impact loads for the impact test rig can easily be added to a finite element model.

It may be that a successful high frequency test rig will confirm the accuracy of one of the predictive methods for finding high frequency dynamic stiffness of baseplates. If this is the case, then a high frequency test may not be required for resilient baseplate Standards. If the predictive models do not correlate well with the high frequency test rig results, then a physical test should be considered for the Standards.

References

[1] Payne, A.R. "Dynamic Behaviour," Chapter 5 in Mernagh, L.R. (ed.) (1968), *Rubber Handbook*, Design Engineering Series, Morgan-Grampian Publishers, Kent/London Payne, 1969, p. 36.

[2] Rail Infrastructure Corporation, *Specification for Vibration Isolation Rail Fasteners* (C3304 v.2.0, Nov. 2001).

[3] German Standard E DIN 45673-1 (1998-07), *Resilient elements used in railway tracks – Part 1: Laboratory determination of static and dynamic characteristics*, Section 6.1.2 ("Dynamic stiffness").

[4] DD ENV 13481-6:2002, *Railway applications – Track – Performance requirements for fastening systems – Part 6: Special fastening systems for attenuation of vibration*, January 2002.

[5] Pandrol® Ltd (1997), "Tests on Studded Rubber Railpads to Establish a Technique for High Frequency Dynamic Stiffness of Railpads", internal company report.

[6] British Standard BS EN 13146-3:2002, *Railway applications – Track – Test methods for fastening systems – Part 3: Determination of attenuation of impact loads*, November 2002.

[7] Hall, M.M. (1968), "The Place of Rubber as an Engineering Material," Chapter 1 in Mernagh, L.R. (ed.) (1968), *Rubber Handbook*, Design Engineering Series, Morgan-Grampian Publishers, Kent/London, p. 9.