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Measurements of the dynamic railpad properties

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Abstract

Numerical models of a railway track rarely take the nonlinear dynamic component behaviour of the railpad into account due to three important problems: first, the considerable amount of calculation time consumed by nonlinear numerical models; second, the lack of validation tools for nonlinear numerical models; third, the lack of data on the nonlinear properties of those components. The ever-increasing power of computers will solve the first problem. Scale models can be a solution for the validation problem. This paper discusses a practical solution for the third problem.

The paper will present an alternative set-up for railpad testing allowing for the measurement of stiffness and damping values between 20 and 2500 Hz with variable preload. It shows the important variation of the dynamic stiffness and the loss factor throughout the chosen frequency range. It is also found that the nonlinear dynamic behaviour of this railway track component certainly affects the dynamic behaviour of the complete track.

1. Introduction

1.1. Railway research at the Vrije Universiteit Brussel

The railway research at the Vrije Universiteit Brussel recently has been focused on the influence of nonlinear component properties on the vertical dynamic behaviour of the track in the frequency range 20–2500 Hz. A nonlinear numerical track model, a laboratory test set-up to measure the dynamic properties of railpads in the frequency range of interest and a one-fifth scaled test track for validation purposes are the tools at hand. This article discusses the laboratory test set-up to measure the nonlinear dynamic properties of railpads and the development of a component model that can be incorporated in the nonlinear numerical track model.

1.2. Importance of nonlinear dynamic behaviour of railpads

The railpad is an important and readily replaceable component of a railway track, as it is the elastic layer between the rail and the sleeper. The railpad has an important influence on the dynamic behaviour of the track. Furthermore, the dynamic behaviour of the track is directly connected to the radiated noise levels and

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the track's wear levels. These phenomena are critical environmental and maintenance issues to be solved in current railway track design.

Typically a railpad is made of rubber, plastic or a composite material such as rubber-bonded cork. Geometry, structure and composition of the railpad are adaptable to certain extents to design a bespoke railpad depending on requirements based on experience or calculation results. It is important to notice that measurements on railpads have shown that the dynamic stiffness properties are nonlinear under different (pre)loading conditions that occur due to the static preloading by the fastening system and the dynamic loading due to wheel passing.

Research on this topic, e.g. by Wu and Thompson [1] and Oscarsson [2], proved that the receptance of the track can partly be optimised towards reduction of noise radiation and track degradation by changing the properties of the railpad. The same research also has shown that the nonlinear dynamic stiffness behaviour of railpads due to loading conditions, etc., certainly affects the tracks behaviour, but the exact extent is not yet fully explored nor validated.

1.3. Dynamic stiffness and damping

The dynamic behaviour of a material or a system can be characterized by its stiffness and damping values. The stiffness and damping values are not constant, but are dependent on the loading frequency, the aging and the loading history, the preload, the temperature, etc. The relations between the dynamic stiffness and damping values and these parameters can be linear or nonlinear dependent on the material or the system at hand.

For the study of vertical vibration, a railpad is usually modelled as a spring and viscous dashpot in parallel, i.e. the Kelvin model. A model of a pad with structural damping with a constant loss factor has also been used and is actually more consistent with the known behaviour of materials such as rubber. Sato [3] has proposed that a rubber pad should be represented by three parameters rather than the conventional two in order to represent the increase in pad stiffness with frequency. Such an improved model, e.g. the Poynting–Thomson model (P–T), evidently has some advantages but it is also more difficult to obtain the necessary parameters.

Although tests can be undertaken in the laboratory, the common opinion is that it is best to extract the unknown parameters from tests in which a vehicle runs over the track containing the rail pads of interest. A compromise is to obtain the parameters from tests in track with an impact hammer or electromagnetic exciter. For both types of test the parameters are found by fitting calculated transfer functions to measured data [4].

The criticism on in situ measurements is twofold. First, the dynamic properties are determined using a numerical model by curve fitting measurements and calculations with the damping and stiffness properties of the railpads as the variable constants. The results are dependent on the numerical model and are only applicable on the measured track. Second, these types of measurements do not take the nonlinear stiffness behaviour of the pads into account. Laboratory measurements are therefore necessary to obtain additional nonlinear information.

2. State of the art of laboratory railpad testers

2.1. General

The laboratory measurement of vibro-acoustic transfer properties of resilient elements is described in Ref. [5]. Three methods are described to measure the dynamic stiffness at different preloads and frequencies: the direct method, the indirect method and the driving point method. The direct method measures the input displacement, velocity or acceleration and the blocking output force. The indirect method measures the transmissibility of an isolator, with the output loaded by a mass/effective mass. Finally, the driving point method measures either the input displacement, velocity or acceleration, and the input force, with the output side of the vibration isolator blocked. Three existing laboratory railpad test set-ups are discussed in this paper. They all use the indirect method.

2.2. Railpad tester TU Delft [6]

The railpad is considered as the only elastic component in the tester, which is a system of tuned masses, preloading springs and the elastic supports, see Fig. 1. A single degree of freedom (sdof) system is sufficient to determine the properties of the railpad based on the output of a single accelerometer, mounted in the centre of the upper suspended mass, and the impact force used for the excitation of the system. The lower mass and elastic supports isolate the measurement from the environment. The dynamic parameters can only be determined at one frequency, the resonance frequency.

2.3. Railpad tester TU-Berlin [7]

Two railpads are placed between steel plates. An adjustable preload acts on the upper steel plate (Fig. 2). The accelerations of the three plates, due to a harmonic excitation, are measured. The stiffness of the railpads can be determined based on those measurements.

2.4. Railpad tester TNO [8]

The measurement apparatus consists of two blocks of known mass between which the resilient element is mounted. The high frequency vibration of the upper block is introduced by using electro-dynamic exciters. Accelerometers are used to measure the vibration of both blocks. The measurement apparatus can be

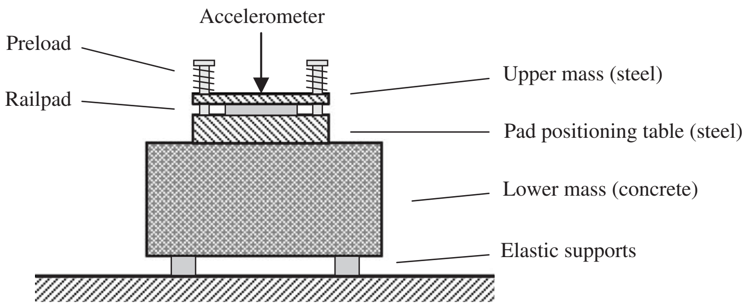

Fig. 1. Test set-up TU-Delft (redrawn from Ref. [6]).

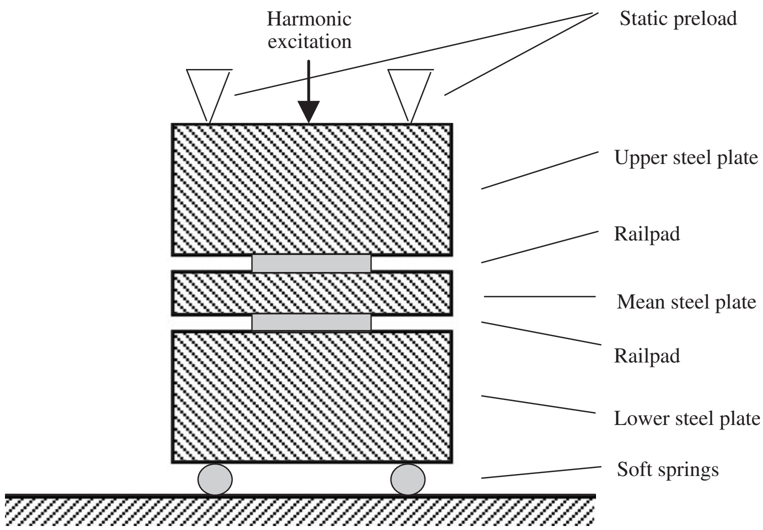

Fig. 2. Test set-up TU-Berlin (redrawn from Ref. [7]).

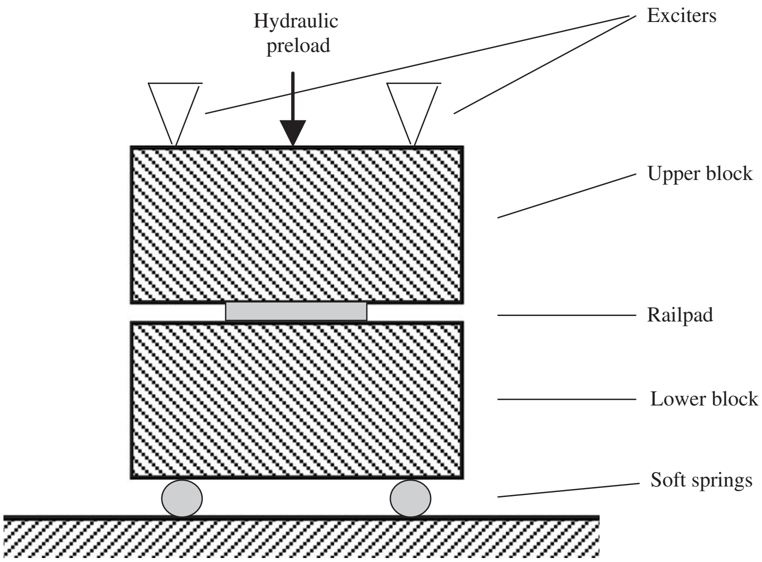

Fig. 3. Test set-up TNO (redrawn from Ref. [8]).

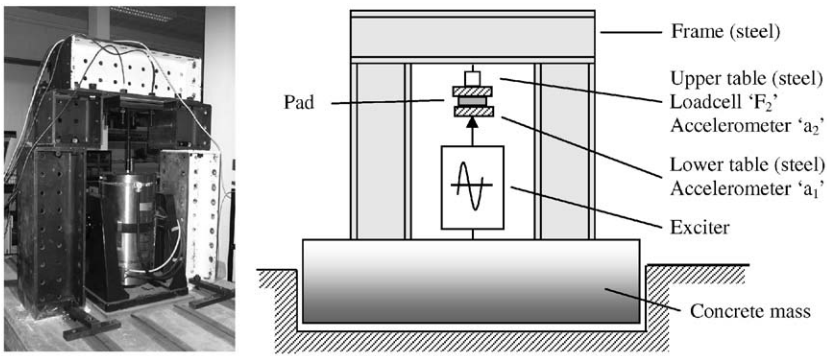

Fig. 4. The alternative elastic pad test set-up.

approximated as a mass–spring system. The stiffness of the railpad can be derived from the equations of motion of the system (Fig. 3).

3. Alternative elastic pad test set-up developed at the VUB

3.1. Description of the alternative elastic pad test set-up

While the previously discussed set-ups make use of the indirect method to measure the dynamic properties of the railpads, the direct method is used for this alternative test set-up (Fig. 4). This has become possible due to the use of a smaller size of test specimen and consequently the lower absolute loading level. The available exciter is powerful enough to generate the combined preload and dynamic load. The whole set-up is inverted to allow the use of a large exciter functioning as the dynamic actuator of the set-up. A frame is build around the exciter. The whole set-up is built on a concrete mass, isolating it from surrounding influences.

The test specimen is centred between 2 steel plates. The top plate is fixed on the frame through a load cell. The bottom plate is connected to the exciter. A DC-signal to the exciter generates the requested preload. The AC-signal of the vibration exciter (Bruël & Kjær System S range, exciter body 4802, big table head 4818) is a sine function with a frequency between 20 and 2500 Hz. The connection between the exciter and the bottom

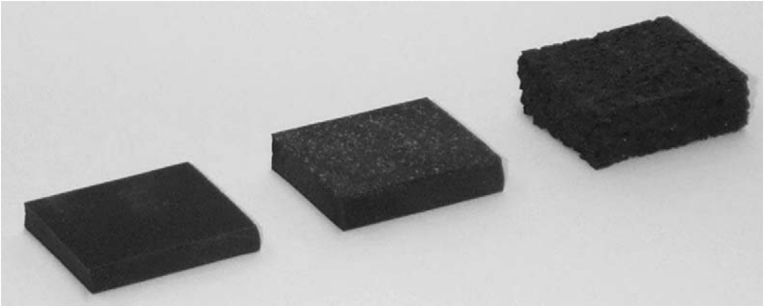

Fig. 5. The test specimens: from left to right the EVA-pad, the DPHI-pad and the SRP-pad.

steel plate is a bar. Two accelerometers are attached to the bottom plate. Signals are captured by a digital data acquisition system, which also allows analysis of the data.

3.2. Description test specimen

The test specimens are cut from railpads to 25 mm × 30 mm. Three railpad types/materials are chosen (Fig. 5). An EVA railpad with a thickness of 4.5 mm is the reference pad. The stiffness as given by the manufacturer is high: $K_{\mathrm{stat}} > 650$ MN/m. The CDM-DPHI-H70 railpad is composed of a layer polyurethane with a thickness of 6 mm and a layer of cork rubber with a thickness of 1 mm. The stiffness properties of the DPHI-pad are: $K_{\mathrm{stat}} = 50$–70 MN/m and $K_{\mathrm{dyn}} < 120$–150 MN/m, as supplied by the manufacturer. The CDM-SRP-L40 is made of CDM-RR or resin-bonded rubber. The stiffness properties of the SRP-pad are: $K_{\mathrm{stat}} > 40$ MN/m and $K_{\mathrm{dyn}} < 100$ MN/m, as supplied by the manufacturer. Each of these values applies to the whole railpad.

4. Measurements

The surface ratio of a railpad and the test specimen is approximately 40 to 1. As an estimation, the same ratio applies for the dynamic stiffness of the railpad and the test specimen with disregard of the shape factor. The measurements are done at preloaded stress levels that are related to those on real-scale railpads. Five preloads are considered: 375, 500, 625, 750 and 1000 N. This is equivalent with preloads on railpads of approximately 15, 20 kN (average preload of the fixation system on the rail), 25, 30 and 40 kN at equivalent stress levels.

The dynamic transfer stiffness $k_{2,1}$ is the frequency-dependent complex ratio of the force F_2 on the blocked output side of a vibration isolator to the complex displacement u_1 or the acceleration a_1 on the input side during simple harmonic vibration:

$$k_{2,1}(f) = \frac{F_2}{u_1} = -(2\pi f)^2 \frac{F_2}{a_1}. \tag{1}$$

The loss factor can be calculated from

$$\eta(f) = \mathrm{Im}\{k_{2,1}(f)\}/\mathrm{Re}\{k_{2,1}(f)\}. \tag{2}$$

The test specimen forms the only transfer path between the excitation and the receiving structure. In practice, there may be mechanical or acoustical parallel transmission paths, which cause flanking transmission. Measurement of the accelerance a_2 in the frequency range of 20–2500 Hz at the top of the frame shows that the flanking transmission is beneath the ISO 10846-2 limits:

$$\Delta L_{12} = L_{a1} - L_{a2} \geqslant 20\,\mathrm{dB} \tag{3}$$

with vibratory acceleration level, $L_a = 20\log(a_{rms}/a_0)$ (dB), a_1 and a_2 the acceleration of, respectively, the input and output flange and a_0 the reference acceleration $10^{-6}\,\mathrm{m/s^2}$.

Figs. 6–9 show the dynamic stiffness and loss factor of an SRP test specimen at different preloads, and of SRP, DPHI and EVA test specimens at a preload of 500 N.

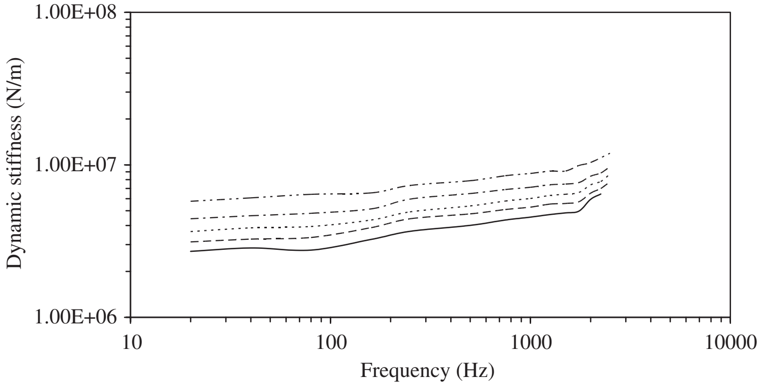

Fig. 6. Dynamic stiffness of SRP specimen at preloads: — 375 N, – – – 500 N, - - - - 625 N, –·–· 750 N and –· ·– 1000 N.

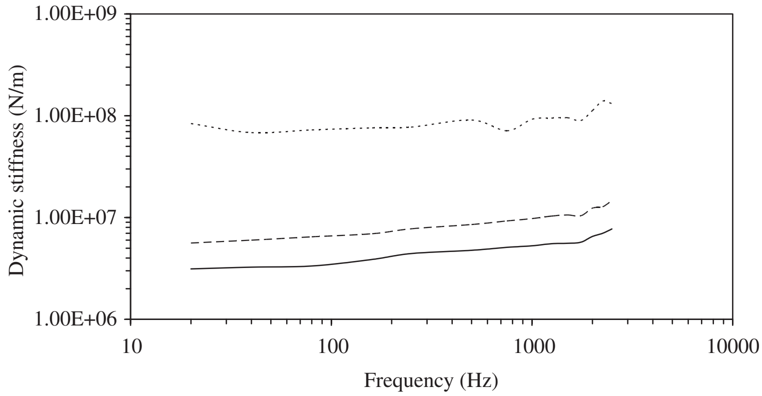

Fig. 7. Dynamic stiffness at 500 N preload of — SRP, – – – DPHI and - - - - EVA specimen.

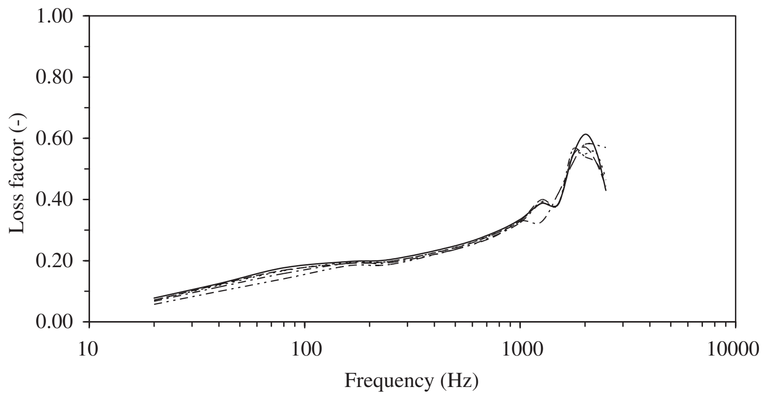

Fig. 8. Loss factor of SRP specimen at preloads: — 375 N, – – – 500 N, - - - - 625 N, –·–· 750 N and –· ·– 1000 N.

The dynamic stiffness of the SRP pad in Fig. 6 increases with frequency. Between 2000 and 2500 Hz, the dynamic stiffness rises more rapidly. The measurements come up to the expectations: higher preloads result in higher dynamic stiffness. The EVA pad is clearly the stiffest and least frequency dependent as seen in Fig. 7. The behaviour of the DPHI pad is similar in comparison with the SRP pad. The measurements do not show noticeable influence of the bi-layered structure of the DPHI pad.

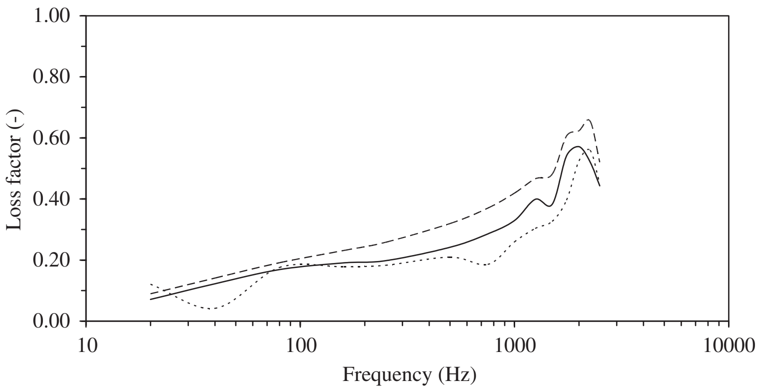

Fig. 9. Loss factor at 500 N preload of — SRP, – – – DPHI and - - - - EVA specimen.

As an indication of the value of the presented test set-up, the measurements can be indicatively compared with those found in literature, taking into account a ratio of 40 between the dynamic stiffness. Thompson et al. [8] presented the dynamic stiffness of railpads up to 1000 Hz at different preloads. The results given here are consistent with those results in that frequency range. The results presented by Knothe et al. [7] up to 2000 Hz are also comparable with the results published here. These comparisons keep in mind the difference in pad material and scale.

The loss factor increases with the frequency as suspected (Fig. 8). However, the measurements suggest an independency with regard to the preload. It is clear that at higher frequencies the measurement of the loss factor is less consistent. This is also shown in Fig. 9 for the three materials tested.

The loss factor of the stiff EVA pad is less coherent, and surprisingly not much lower than the loss factor of the SRP and the DPHI pads. The DPHI pad has the highest loss factor.

5. Material model

5.1. Introduction

A suitable material/component model of the railpad is needed for the use in a numerical modelling program. Due to the demonstrated dependence of the dynamic stiffness and damping on the frequency and the preload, a dedicated material/component model is required. The standard Kelvin model (spring and damper in parallel) is generally used in numerical models for the mechanical modelling of a one-dimensional elastic component. When the component shows a frequency dependency, the P–T model (spring parallel to a damper and spring in series) is used e.g. by Ripke [9]. Fig. 10 shows the mechanical modelling and the frequency dependencies of the stiffness and the damping. De Man [10] adapted the P–T model further in order to separate the influences of frequency from the influences of preload. Table 1 indicates the mathematical representation of the models.

5.2. Comparison between adapted P– T material model and measurements

Figs. 11 and 12 show the measurement results on SRP test specimen in comparison with predictions of the adapted P–T material model at different preloads. The values are obtained with the least-squares method where the difference between the calculated P–T data and the measured data is minimized. Similar results were found for the other test specimens.

The dynamic stiffness predicted by the adapted P–T material model is in good agreement with the previously described measurement results (Fig. 11) up to 2000 Hz. The increase in stiffness above 2000 Hz cannot be mirrored by the P–T model.

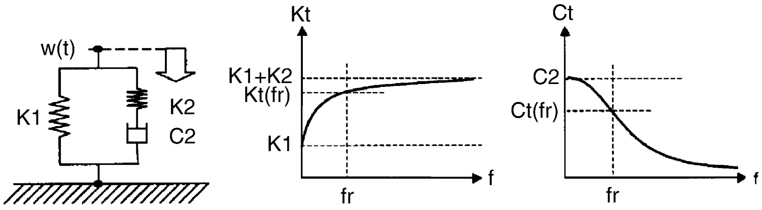

Fig. 10. Poynting–Thomson mechanical models and frequency-dependent behaviour [9].

Table 1
Mathematical representation of the Poynting–Thomson model [9]

Property	P–T model	Adapted P–T model
K_t	$K_1 + K_2\alpha$	$K_1\beta_1 + K_2\alpha\beta_2$
C_t	$C_2\alpha(z^2/n^2)$	$C_2\alpha(z^2/n^2)\beta_3$

K_t, (total) stiffness (N/m); C_t, (total) damping (Ns/m); K_1, stiffness; frequency independent (N/m); K_2, stiffness frequency dependent (N/m); C_2, viscous damping frequency dependent (Ns/m); α, coefficient $\alpha = (n^2/n^2 + z^2)$ (–); n, radian frequency $n = 2\pi f$ (s^{-1}); z, partial inverse loss value $z = (K_2/C_2)$ (s^{-1}); $\beta_{1,2,3}$, preload coeff. $\beta_{1,2,3} \in \{(1 + P/P_0)^x, 1\}$ (–); P, preload (N); P_0, reference preload (N); x, exponential preload influence (–).

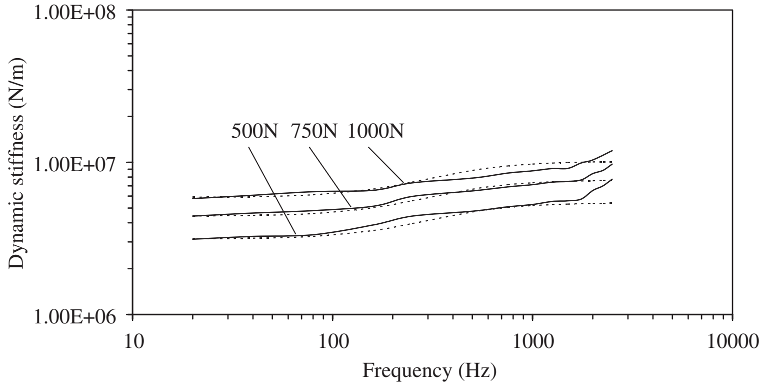

Fig. 11. Dynamic stiffness of SRP specimen at preloads 500, 750 and 1000 N: — measurement; ---- adapted P–T material model.

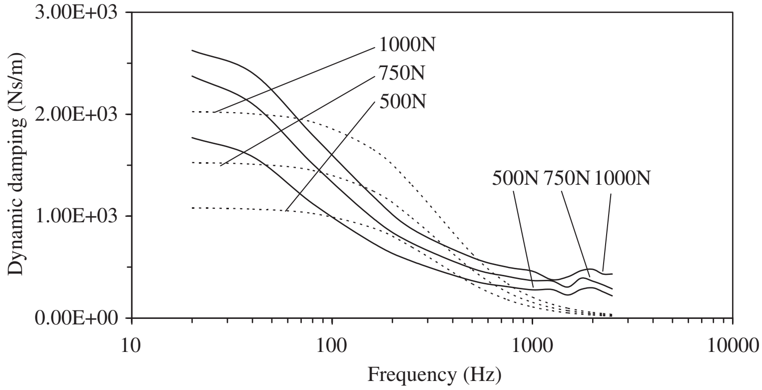

Fig. 12. Dynamic damping of SRP at preloads 500, 750 and 1000 N: — measurement; ---- adapted P–T material model.

The equivalent viscous damping C can be calculated from the loss factor

$$\eta = \frac{nC}{k} \tag{4}$$

with the radian frequency n and dynamic stiffness k.

Fig. 12 shows that the dynamic damping behaviour of the SRP specimen cannot be satisfactorily described by the adapted P–T model. By changing the P–T parameters, the resemblance between model and damping measurements can be improved, but consequently the good modelling of the dynamic stiffness is lost.

6. Conclusion

The dynamic properties of elastic pads are measured in a frequency range of 20–2500 Hz at different preloads. These values can be used as input data in nonlinear numerical models of a railway track. The availability of independent laboratory test methods is essential for the success of dynamic railway track design software.

The proposed test set-up in this paper uses the direct measurement method while most papers commonly use the indirect method, e.g. TU Delft, TNO, TU Berlin. The application of the direct method is possible due to a smaller test specimen and a powerful exciter. It offers a fast and efficient alternative measurement system.

The measurements show the influence of railpad type, frequency and preload on dynamic stiffness and loss factor. The railpad materials tested are EVA, DPHI and SRP, with EVA the hardest and SRP the softest. The influence of the preload and frequency is clearly nonlinear.

It is shown that the adapted P–T material model can be used for the dynamic stiffness up to 2000 Hz. However, the damping is not accurately predicted and needs further attention.

Acknowledgements

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