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Influence of non-linear stiffness and damping on the train-bridge resonance of a simply supported railway bridge

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ABSTRACT

Previous experimental work has identified variations in the natural frequency and the modal damping ratio of the first vertical bending mode of vibration of a simply supported, single span steel–concrete composite bridge. It was found that the natural frequency decreased and the modal damping ratio increased with increasing amplitudes of vibration. This paper illustrates the influence of these variations on the train-bridge resonance of this particular bridge by means of a non-linear single degree of freedom system, based on the previously mentioned experimental results. As one might expect, the results indicate that the influence of the increasing damping ratio leads to a considerable decrease in the resonant amplitude whilst the decreasing natural frequency decreases the critical train speed at which resonance occurs. Further studies along this line of research may help us reduce the uncertainties in dynamic assessments of existing bridges based on dynamic measurements and improve our understanding of the dynamic properties of railway bridges in general.

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1. Introduction

The dynamic response of railway bridges subjected to high-speed trains is mainly governed by different states of resonance between the bridge and the train. The load induced by the train will, at certain train speeds, have components with frequencies that match eigenfrequencies of the structure. Thus, many possible combinations of train configurations and train speeds can exist which cause states of train-bridge resonance. This is one of the main issues in design of new bridges for high-speed railway lines and in dynamic assessments of existing bridges, which is becoming increasingly interesting for railway owners who wish to increase the maximum allowed train speed and axle loads.

From an analysis of single degree of freedom systems, one knows that in a state of resonance, the amplitude of vibration is mainly governed by the damping of the system. This also holds true for multi degree of freedom systems as well as for continuous systems, although in such cases, different combinations of modes may be relevant. However, variations in the eigenfrequency are also likely to influence the state of resonance, mainly by altering the critical train speed.

Previous studies [2,4] have given indications that for certain bridges, the damping ratio and the natural frequency have a dependency on the amplitude of vibration. The nature of these non-linearities are not well known but candidates have been suggested in the non-linear material properties of soil materials and concrete, which both have the same tendency: the damping increases and the stiffness decreases with the deformation of the materials. Fink and Mähr [3] reported experimental findings from a scaled laboratory model of a ballasted railway bridge which support the hypothesis that the ballast is one of the main sources to this behavior.

This paper aims at illustrating the influence of the non-linear dynamic properties of a simply supported, ballasted composite bridge on its response at resonance. A simple single degree of freedom system representing the first vertical bending mode of the bridge is used to simulate the response caused by a typical freight train and for a theoretical study of the train bridge resonance based on the Eurocode HSLM (*High Speed Load Model*) trains [5].

As the state of resonance is often dominated by a single mode of vibration, a qualitative analysis of this type of models may provide some insight into the real behavior at resonance. A formalized knowledge of the railway bridge response at resonance may lead to substantial savings for society, if it turns out that at resonance, the increased damping leads to a much smaller response than that predicted by linear theories.

2. Theory

2.1. A non-linear single degree of freedom system

Without explicitly knowing the sources to the non-linear behavior, models can only be devised in a "black-box" sense.

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Measurements can give estimates of the relations between the dynamic properties (natural frequency and damping ratio) of different modes of vibration and the amplitude of vibration in those modes, see [4] and the references therein. Given such relations, a non-linear single degree of freedom system can be established

$$m\ddot{x} + c(x)\dot{x} + k(x)x = f \tag{1}$$

where x is the generalized coordinate of the fundamental mode of vibration, m is the generalized mass, $c(x)$ is an amplitude dependent viscous dashpot coefficient, $k(x)$ is a non-linear spring constant and $f = f(t)$ is the generalized forcing function.

The non-linearities are assumed to be small in the sense that the frequency does not vary much around the limit value at $\ddot{x} = 0$ and the mode shape is assumed to be constant, independent on the amplitude of vibration. Furthermore, the ''black-box'' nature of the proposed model presumes that variations in support stiffness and damping and the interaction between the structure and the embankments and with the track superstructure are all grouped together in the eigenfrequency and the damping ratio.

The load model used to define the generalized force function $f(t)$ is also subjected to some simplifying assumptions, namely that the train-bridge interaction may be neglected, thus leaving out the variation in mass damping and perhaps to some extent in stiffness, caused by the passing train.

Frybá [1] derived a solution for the response of a simply supported beam subjected to a pulse train moving along the beam. In the present analysis, we wish to solve for the temporal coordinate using a numerical technique in order to include the non-linear system parameters, but the generalized force for the first mode of vibration is approximated in the same way as in [1]:

$$f(t) = \sum_{i=1}^{N} F_i \epsilon_i(t)\phi(ct - d_i) \tag{2}$$

where F_i is the axle load of axle number i, N is the number of axles, $\epsilon(t)$ is a function defined by

$$\epsilon_i(t) = H(t - d_i/c) - H(t - (d_i + L)/c) \tag{3}$$

where $H(t)$ is Heaviside's function. Furthermore, d_i is the distance from the ith axle to the first point on the beam, L is the length of the beam and $\phi(x)$ is the first (vertical bending) mode of vibration

$$\phi(x) = \sin\left(\frac{\pi x}{L}\right) \tag{4}$$

This representation of the load function is a consequence of expanding the spatial coordinate in a Fourier series. This series will repeat itself indefinitely along the spatial coordinate, but we are only interested in $x \in (0, L)$. The function $\epsilon_i(t)$ simply ensures that the load is not applied to the repeated occurrences of the physical structure. Otherwise, the analysis would comprise a structure equivalent to an infinite continuous beam on simple supports.

The parameters of Eq. (1) can be rearranged so that the following equation is obtained

$$\ddot{x} + 2\xi(x)\omega_n(x)\dot{x} + \omega_n^2(x)x = \frac{f}{m} \tag{5}$$

where $\xi(x)$ and $\omega_n(x)$ are the amplitude dependent damping ratio and natural circular frequency of the fundamental mode of vibration.

A methodology to determine the amplitude dependency of the natural frequency and damping ratio from measured free vibrations after the passage of a train using the continuous wavelet transform (CWT) have been presented in [4]. More general applications of the CWT have been presented in several papers, see for example [6]. In this paper, bilinear relations based on the analysis presented in [4], were used to model the amplitude-dependency of the dynamic properties of the first mode of vibration. These relations, which may be represented by the generic relation

$$g(\ddot{x}) = \begin{cases} g_0 + k|\ddot{x}|, & |\ddot{x}| \leqslant \ddot{x}_c \\ g_c, & |\ddot{x}| > \ddot{x}_c \end{cases} \tag{6}$$

are shown in Fig. 1 with the parameters given in Table 1. The linear part of these functions were determined by means of the CWT, based on five train passages at slightly different speeds. For further details, the reader is referred to [4]. The constant parts of these functions have been assumed and reflect the lack of knowledge about these relations at accelerations greater than 0.3 m/s^2.

The relations given by Eq. (6) were derived using measurements of acceleration. However, in solving non-linear differential equations it is much more convenient to have the non-linearity on the displacement and/or the velocity as then, well-known numerical methods may be directly applied. In the present context, this does not pose any serious difficulties, because the non-linear relations given by Eq. (6) are defined using the free vibrations of a single mode of vibration. Therefore, the displacement during the free vibrations can be determined from measurements of acceleration simply by applying a high-pass filter to the signal and integrate it numerically. By doing so, and normalizing the results it is easy to verify that the displacement during free vibrations is proportional to the accelerations according to

$$u(t) \sim \frac{\ddot{u}(t)}{\omega^2} \tag{7}$$

with a phase-shift π. This is shown in Fig. 2, where ω was taken as $2\pi 3.9$ rad/s which clearly shows that the variation in frequency is slow enough to make this approximation feasible, i.e. the acceleration of the free vibrations may be obtained from the displacement function by applying a scaling and a translation. Formally, this means that the frequency and amplitude modulated signal which is considered here

$$u(t) = A(t)\cos(\omega(t)t) \tag{8}$$

has the property that $\dot{\omega}(t) \ll 1$, i.e. slow variations in $\omega(t)$. Thereby, the generic relation (6) can be restated as a function of displacement, simply by making the change of variables $\ddot{x} = -\omega_0^2 x$, with ω_0 being the natural frequency at small amplitudes of vibration.

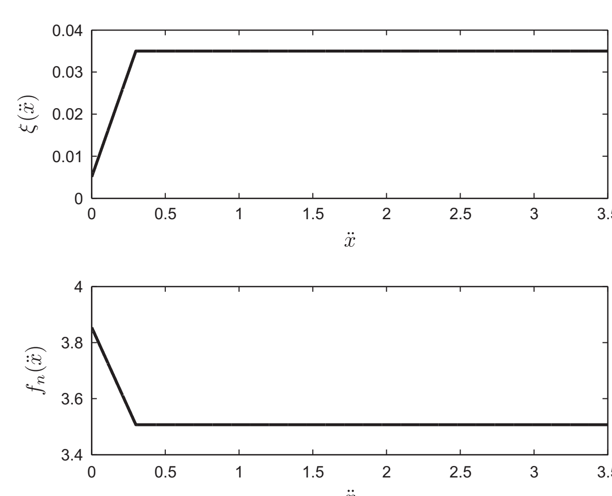

Fig. 1. Top: The damping ratio function (in %) for the SDOF-model. Bottom: The frequency function (in Hz) for the SDOF-model.

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