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Application of an optimal wavelet transformation for rail-fastening system identification in different preloads

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ABSTRACT

A new methodology for the modal parameter identification of a rail fastener system is proposed based on the optimal wavelet transformation. The method namely the conditional entropy criterion is proposed and the Morlet wavelet transformation is optimally tuned for the case study. It is shown that one may reach to desired frequency resolution to separate the mode shapes and simultaneously arrive at the necessary time resolution to mitigate the end effects and properly estimate the connection damping. An experimental set-up has been provided and different tests have been arranged in different fastening conditions. A parametric study on the influence of the fastening torque on the modal parameters is provided. It is proved that this new wavelet-based strategy can accurately estimate the modal frequencies and damping of such a nonlinear case.

1. Introduction

The function of a railway infrastructure is generally to support and transmit the wheel loads from rail to the railway foundation with a reduction in pressure. Between different parts of the railway infrastructure, rail fastening system is one of the key players in railway noise generation. It is also a key element in providing consistent contact between the rail and sleepers. Due to their important role, these elements are usually inspected in regular maintenance inspections. Estimation of the corresponding modal properties such as the clip stiffness and damping has been of interest due to their direct contribution in railway noise generation particularly for the slab tracks in which most of the sound energy dissipation takes place in the fastening system and the rail-pad.

Although the dynamic behavior of the rail-pads has been studied and well addressed in the literature [1,2] there exists very few publications on the matter of rail fastening systems because of (I) very high sensitivity to the preload (II) diversity of the clips and (III) complexity of their dynamic behavior. Thompson and Verheij [3] employed classical modal analysis in the frequency range of 100–1000 Hz to determine stiffness and damping of the NS type clamps in the Netherland railways. A comprehensive survey has been presented by DeMan [4] which is reviewing dynamic properties of the railway track elements. Static stiffness and stress analysis of SK type clips has been studied using the Finite Element Method by Lakusic and Bartos [5].

It has been addressed in the literature that, in sensitive situations such as our case study in which the dynamic properties are very dependent on the preload, classical modal analysis has remarkable error and is not recommended for such cases. The earliest research done by Staszwski [6], proposed the wavelet transform for identification of MDOF non-linear systems. The ridges and skeletons of the wavelet transform are used to obtain the instantaneous envelope and the frequency characteristics. These characteristics are employed to obtain backbone curves and decay envelope of the system which in turn

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are used for the system identification. Lamarque et al. [7] introduced a similar logarithmic decrement formula to estimate damping in a MDOF system based on the wavelet transform applied to a filtered component of the original response. Identification of the nonlinear systems using the continuous wavelet transform has been proposed by others including Lardies et al. [8], Ta et al. [9].

The wavelet transform has been proved to be very suitable tool to estimate the modal parameters for non-stationary dynamical forces and it is a powerful method for analysis of the mechanical system excited by random forces and structures subjected to the ambient loading [10–13].

One of the important challenges in modal parameter identification is how to select the optimal mother wavelet function. Wavelet function should be able to capture both the time and frequency information of the signal. In other words, the appropriate wavelet transform should provide a tradeoff between the time and frequency resolution. The optimal wavelet transformation should provide good frequency resolution to be able to separate different close modes and it needs good time resolution for accurate estimation of the damping and suppressing the end effect [14].

The Shannon entropy [15] measures the energy concentration or uncertainty and gives a criterion for selection of the basic wavelet. An entropy criterion has been used by Lardies and Ta [9] to choose the best Morlet wavelet parameter for the modal identification. Determination of the optimal wavelet shape using the entropy concept has been addressed by other researchers as well. Park et al. [16] proposed an effective method for extracting the beat characteristics and modal damping ratios of the bell type structures using the continuous wavelet. Other references [17,18] have used entropy criterion to select the optimal wavelet transformation. Important note here is that, based on the application, researchers so far have used a single aspect of the wavelet coefficient for the entropy computation. Users normally employ an entropy criterion for which the optimal wavelet approaches to appropriate frequency resolution or fitting time resolution dependent on the case.

In the present paper, the vibration behavior of a rail fastening system is theoretically and experimentally analyzed using an optimal wavelet transformation. Since the rail fastening properties are frequency and time dependent, so the wavelet transformation is an appropriate tool for identification of the modal parameters. A new methodology namely the conditional entropy criterion is proposed to optimally tune the Morlet wavelet transformation fit to our case study. It is experimentally shown that the proposed dynamical model can accurately predict the characteristic behavior of the system. A parametric study is then carried out and effects of the fastening torque on the modal parameters of the system are investigated.

2. Mathematical modeling

Based on the schematic representation of the rail-fastener-sleeper system shown in Fig. 1-I, the dynamic characteristic behavior of the system may be represented by a two-degree-of-freedom arrangement as shown in Fig. 1-IV. In the model of the rail fastening system, rail is modeled as a lumped mass, bolts and washer as the springs with stiffness k_b and k_w respectively and clips as a rigid beam and a spring with equivalent stiffness k_b as shown in Fig. 1-II. Equivalent springs of the fastening bolt and clip are arranged in series since the free end of the bolt spring (point A) is attached to the clip spring at point B and the deflection is exerted on the clip at point C. The equivalent subsystem is shown at the Fig. 1-III. The consequent stiffness is given by

$$k_e = \left(\frac{a}{b}\right)^2 \frac{k_b k_c}{k_b + k_c} \tag{1}$$

in which a and b are the two arms represented in Fig. 1, k_c is equivalent stiffness of the clip and washer, ΔF and Δp are the preloads applied to the bolt and the rail respectively so that

$$\Delta p = -(a/b)\Delta F \tag{2}$$

In addition, in Fig. 1, m_1 and m_2 respectively denote the rail and base-plate mass and k_1 and k_2 represent the rail-pad and the base-plate pad stiffness. De Man model which is called AP-T model is applied for mathematical description of the pad stiffness and damping [4]. The stiffness and damping of the rail pads are both assumed to be functions of the excitation frequency and the exerted pre-load

$$K_t = K_1\beta_1 + K_2\alpha\beta_2 \tag{3}$$

$$C_t = C_2\alpha(z^2/\omega^2)\beta_3 \tag{4}$$

in which K_t (N/m) denotes the total stiffness; C_t (Ns/m) total damping; K_1 (N/m) frequency independent stiffness; K_2 (N/m) frequency dependent stiffness; C_2 (Ns/m) viscous damping frequency dependent; α coefficient $\alpha = (\omega^2/\omega^2 + z^2)$; ω angular frequency $(\omega = 2\pi f)$; z partial inverse loss value $(z = (K_2/C_2))$; $\beta_{1,2,3}$ preload coefficient $\beta_{1,2,3} \in \{(1 + P/P_0)^x, 1\}$; P (N) preload; P_0 (N) reference preload and x, exponential preload influence. It is theoretically seen that by increasing the frequency, the consequent damping decreases (see Eq. (4)) and the equivalent stiffness increases (see Eq. (3)).

Experimental data indicates that influence of the pre-load on the damping value is in contrary with the influence of the frequency [4]. It has been proved that the stiffness and damping properties of the rail-pad are both frequency-dependent. For new rail pad materials, the stiffness value is an increasing function of the frequency while the damping value decreases with any increases in excitation frequency.

Corresponding time-dependent equation of motion of the time-varying system can be summarized in the form of

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{y}_1(t) \\ \ddot{y}_2(t) \end{bmatrix} + \begin{bmatrix} c_1(t) + c_2(t) & -c_2(t) \\ -c_2(t) & c_2(t) \end{bmatrix}\begin{bmatrix} \dot{y}_1(t) \\ \dot{y}_2(t) \end{bmatrix} + \begin{bmatrix} k_1(t) + k_2(t) & -k_2(t) \\ -k_2(t) & k_2(t) + k_3 \end{bmatrix}\begin{bmatrix} y_1(t) \\ y_2(t) \end{bmatrix} = \begin{bmatrix} f_1(t) \\ f_2(t) \end{bmatrix} \tag{5}$$

in which, coefficients $c_i(t)$ and $k_i(t)$ for $i = 1,2$ are defined to be the convolution operator. If they get applied to the functions $\dot{g}(t)$ and $g(t)$, we have

$$c_i(t)\dot{g}(t) = c_i(t) * \dot{g}(t) = \int_0^t c_i(t-\tau)\dot{g}(\tau)d\tau \quad i = 1,2 \tag{6}$$