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Research on Synthetic Spectrum for Time-varying Vibration Loads of Railway Vehicles

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Abstract

The durability of equipment mounted on railway vehicles within the expected life is usually tested based on IEC 61373 standard. However, the frequency characteristics and the instantaneous high-amplitude loads of actual time-varying loads cannot be expressed by the load spectrum that given in the IEC 61373 standard, which is significant for the vibration fatigue analysis of the equipment. Therefore, a synthetic method is proposed to obtain the load spectrum that has the has equal damage potential with time-varying vibration loads, based on the fatigue damage spectrum solved in the time domain. Then the applicability of synthetic spectrum for vibration fatigue analysis is verified by measured dynamic stress. The results show that damage evaluated by the synthetic spectrum obtained with the proposed method is consistent with the damage evaluated the measured stress. In additional, the amplitude of synthetic spectrum for the time-varying loads in a maintenance cycle exceeds the standard spectrum at some frequencies, which facilitates accurate assessment of equipment fatigue damage.
Keywords: railway vehicles equipment, time-varying loads, damage consistency, synthetic spectrum.

1 Introduction

Many equipment is mounted on the main structure of railway vehicles to meet the needs of operation and maintenance, such as the flange lubrication equipment is mounted on bogie frame to reduce the wear of wheel-rail [1]. The equipment is subjected to random vibration loads due to the irregularity of wheel or track in service.

At present, the durability of equipment within the expected life is usually tested based on IEC 61373 [2]. However, in recent years, fatigue failure of equipment has occurred frequency [3-5]. The reason is pointed out in [4] that the vibration load given in IEC 61373 cannot characterize the frequency characteristics of the actual time-varying loads, which is especially significant for structural vibration fatigue analysis.

Xie [5] characterized the frequency characteristics of the measured stationary Gaussian load of the bogie by the power spectral density (PSD), and then evaluated the fatigue damage of the brake pipe equipment mounted on the bogie. But the time-varying (i.e. non-stationarity) loads caused by the diversity track and the evolution of wheel or rail irregularity is not considered. The Miner's rule shows that the damage is exponentially related to the stress amplitude, which means that if the instantaneous high-amplitude loads generated by the rail join, serious rail wear area, etc are ignored, the estimated lifetime will have a large error with the actual value [6]. The MIL-STD-810F standard [7] provides a synthetic method for time-varying vibration loads in a long-term service environment based on the theory of damage consistency, the inverse of fatigue damage spectrum (FDS) [8]. In [9], the synthetic spectrum for the profiles of time-varying vibration loads of the automobile coil spring developed according to the road conditions was obtained with the inverse of FDS, but the non-stationarity of the loads in each profile was ignored.

Firstly, the time-varying vibration loads of equipment mounted on bogie frame is measured and synthesized based on the inverse of FDS in this paper. Then, the validity of the synthetic spectrum is verified by the measured dynamic stress. Finally, the synthetic spectrum that has equal damage potential with the time-varying loads in a maintenance cycle is obtained.

2 Methods

The sensors installed on flange lubrication equipment are shown in Figure 1, in which the vibration acceleration was measured three times and the dynamic stress was measured once for the whole track line in a maintenance cycle.

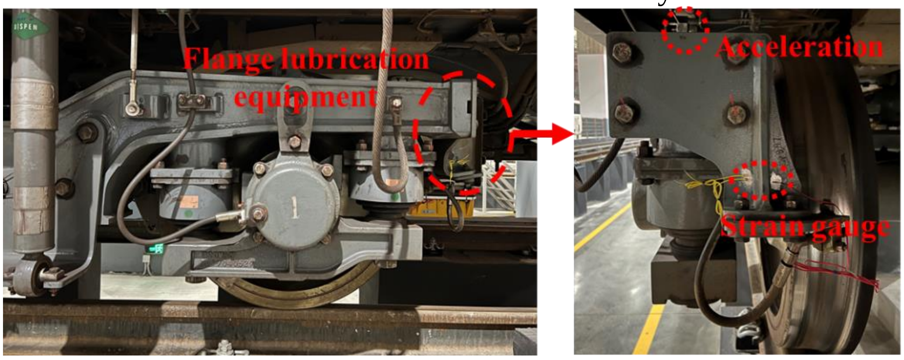


Figure 1: Field test.

The fatigue damage spectrum (FDS) is proposed to evaluate damage potential of the vibration load, as shown in Figure 2.

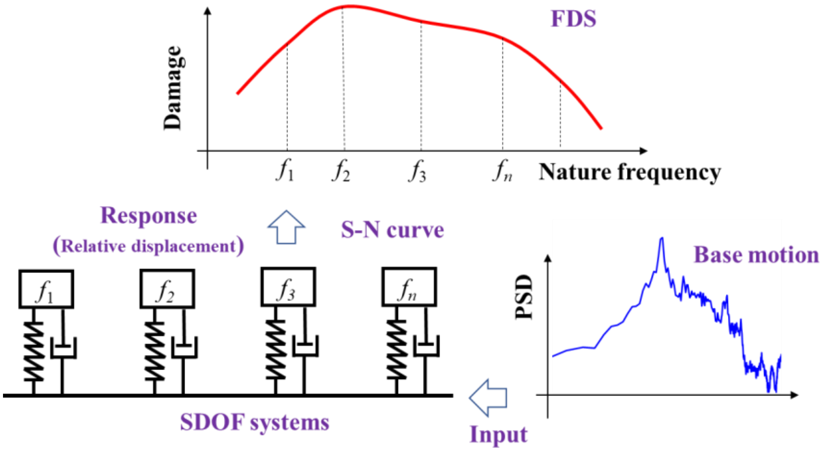


Figure 2: The process for solving FDS.

(1) A set of single degree of freedom (SDOF) systems are developed with a fixed damping ratio ξ and different natural frequencies f_n.

(2) The vibration loads PSD G_x is used as base motion to calculate the response of each SDOF system in the form of relative displacement PSD, which multiplied by the square of elastic modulus E^2 to obtain the stress PSD.

(3) The probability density function of stress peaks S_p represented by the stress PSD conforms to the sum of Gaussian and Rayleigh distribution:

$$p(S_p)=\frac{\sqrt{1-r^2}}{z_{rms}\sqrt{2\pi}}e^{-\frac{(S_p/E)^2}{2(1-r^2)z^2{}_{rms}}}+\frac{(S_p/E)r}{2z^2{}_{rms}}e^{-\frac{(S_p/E)^2}{2z^2{}_{rms}}}\left[1+erf\left(\frac{(S_p/E)r}{z_{rms}\sqrt{2(1-r^2)}}\right)\right] \quad (1)$$

$$erf(\chi)=\frac{2}{\sqrt{\pi}}\int_0^{\chi}e^{-\lambda^2}d\lambda\,;\, z_{rms}=\sqrt{\frac{QG_x(f_n)}{4(2\pi f_n)}} \quad (2)$$

where, erf is the error function; r is spectral irregularity factor; z_{rms} is the rms of the relative displacement; Q is quality factor. Therefore, the cycles of stress peak are counted by:

$$n=v_pT\int_{S_p}^{\infty}p(S_p)dS_p \quad (3)$$

where, v_p is frequency of peaks; T is the duration of the vibration load.

(4) Based on the S-N curve and Miner's rule, the damage of each SDOF system can be calculated and the FDS is obtained:

$$FDS(f_n)=\frac{n}{N}=\frac{E^m}{C}f_nT\left(\frac{QG_x(f_n)}{2(2\pi f_n)^3}\right)^{m/2}\Gamma(1+\frac{m}{2}) \quad (4)$$

where, Γ is the Gamma function; C, N and m are the parameters of S-N curve, $S_p^mN=C$.

For the vibration loads experienced by railway vehicles in service, the equal damage synthetic spectrum can be obtained with the inverse of FDS:

$$G(f_n) = 2\frac{(2\pi f_n)^3}{Q}\left(\frac{\sum FDS_i(f_n)C}{f_n T_S E^m \Gamma(1+\frac{m}{2})}\right) \quad (5)$$

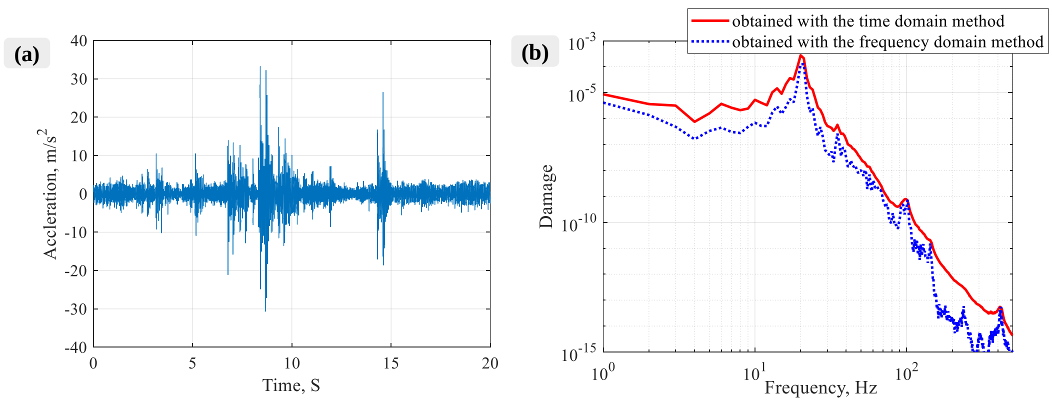
where i represents the number of vibration load conditions.

Eq. (5) is applied to synthesize stationary Gaussian vibration loads that can be represented by the PSD. But for time-varying loads, the PSD represents the average intensity, so the synthetic spectrum obtained with Eq. (5) cannot contain high-amplitude loads which are significant for fatigue assessment [6]. Therefore, the time domain method is proposed to consider high-amplitude loads. i.e. the time-varying load is directly used as base motion to calculate the relative displacement response $z(t)$ of each SDOF system with the Duhamel integral (Eq. 6). The stress is calculated by elastic modulus and counted by the rainflow method. The damage is then evaluated to obtain the FDS and the synthetic spectrum is obtained with Eq. (5).

$$z(t) = -\frac{1}{2\pi f_n}\int_0^t x(\tau)e^{-\xi 2\pi f_n (t-\tau)}\sin(2\pi f_n(t-\tau))d\tau \quad (6)$$

3 Results

The FDSs and synthetic spectra for the time-varying vibration loads are obtained with the frequency domain method (Eq. (1) ~ Eq. (5)) and the time domain method respectively, as shown in Figure 3. The distribution of peak amplitudes of the two synthetic spectra is similar. When solving the FDS by the time-domain method, the instantaneous high-amplitude loads can be considered, by which synthetic spectrum obtained exhibited a higher intensity than the synthetic spectrum obtained with the frequency domain method, approximately 1.5 times. In addition, the black line used for comparison in the Figure 3(c) presents the vibration level of bogie given by the IEC 61373 standard.



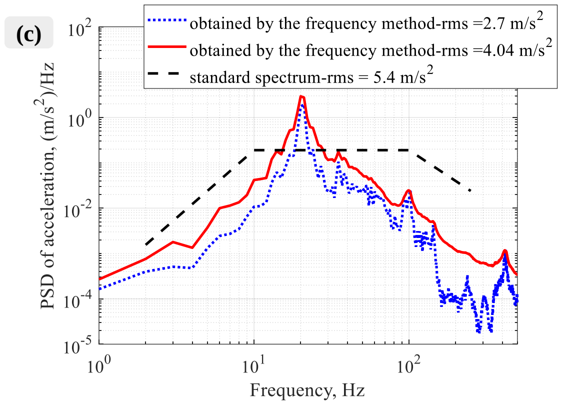


Figure 3: (a) Time-varying loads; (b) FDS; (c) synthetic spectrum.

The synthetic spectra are used as loads to calculate the stress PSD response by the finite element model (FEM) of flange lubrication equipment. Then the stress cycles counted by Dirlik model [6] and damage evaluated by the IIW standard [5] are compared with results calculated by measured stress, as shown in Figure 4. It can be seen that compared with the other spectra, the cycles of stress amplitude represented by stress response PSD calculated by the synthetic spectrum obtained with the time domain method is closest to the cycles of measured stress, and the error of damage is only 28.3%.

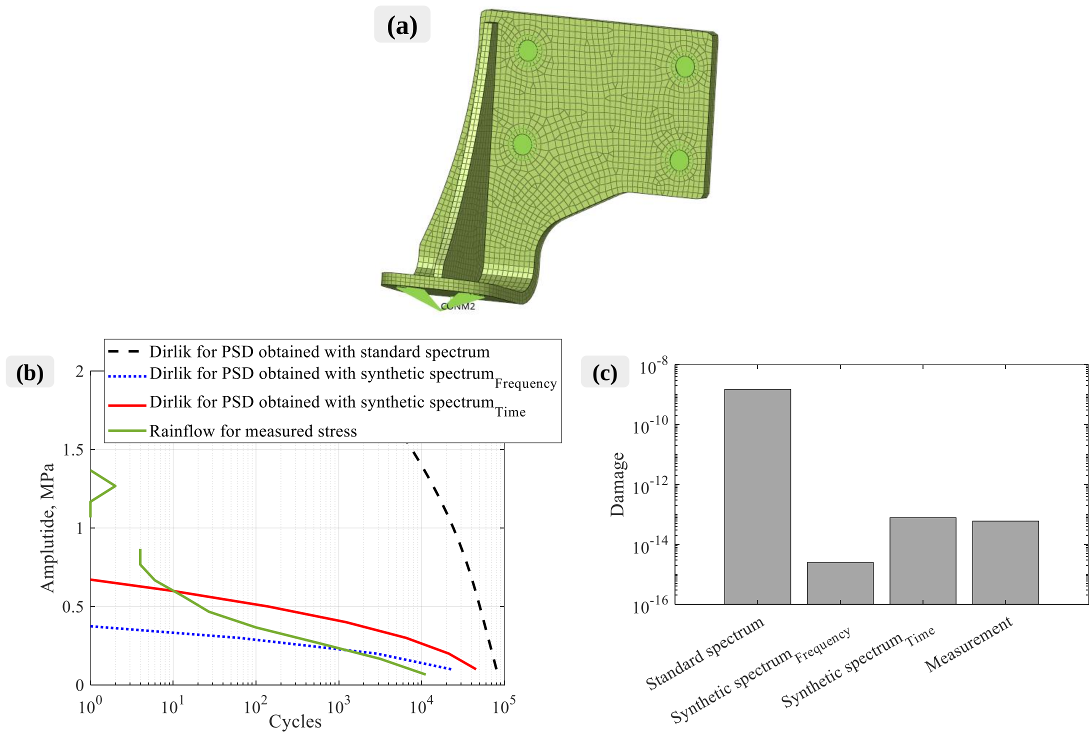


Figure 4: (a) FEM; (b) Cycles of stress amplitude; (c) damage.

The time-varying loads of equipment in a maintenance cycle is synthesized with the time domain method, as shown in Figure 5. The rms of the synthetic spectrum is less than the rms given in the IEC 61373 standard. However, the synthetic spectrum can characterize the frequency characteristics of time-varying vibration loads and exceeds the amplitude of standard spectrum at some frequencies (for Z-direction: 10 ~ 30, 50 ~ 100 Hz; For Y-direction: 65-100 Hz).

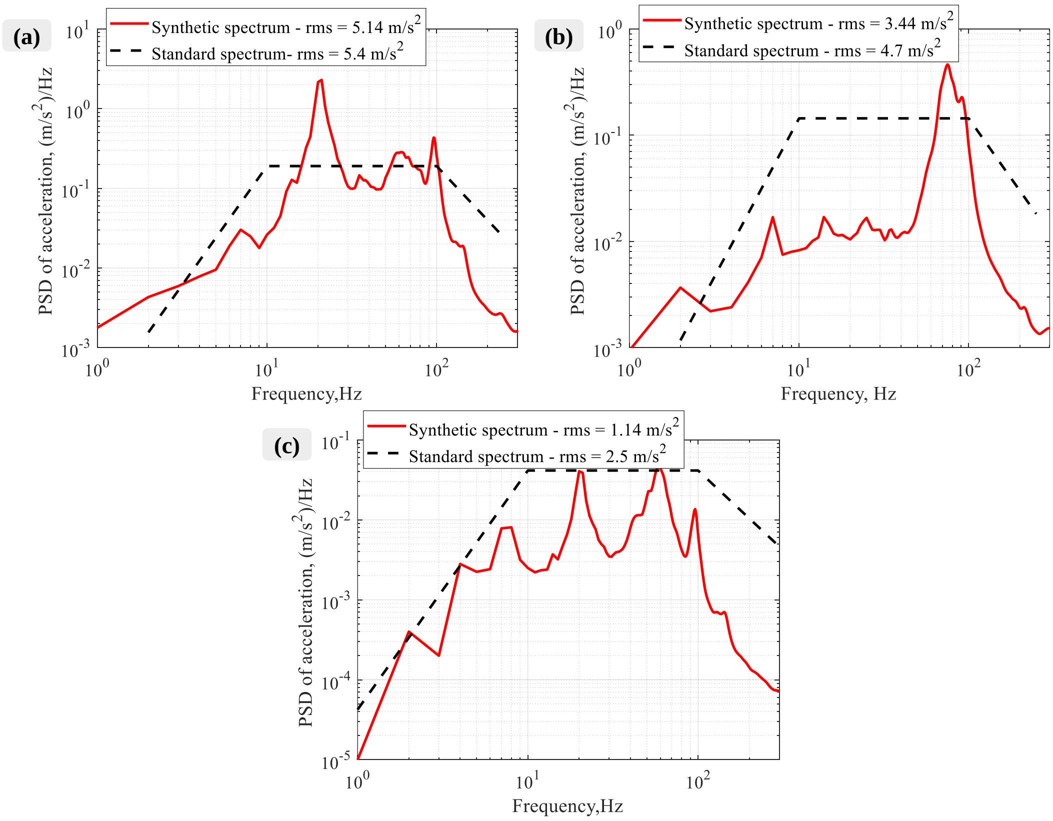


Figure 5: Synthetic spectrum for time-varying loads in (a) Z-direction; (b) Y-direction; (c) X-direction.

4 Conclusions and Contributions

The synthetic spectrum obtained with the proposed method can not only express the frequency characteristics of time-varying loads, but also contain instantaneous high-amplitude loads, by which the damage evaluated is closest to damage evaluated by the measured stress.

Compared with the IEC 61373 standard, the amplitude of synthetic spectrum for the time-varying loads of equipment in a maintenance cycle exceeds amplitude of the standard spectrum at some frequencies, which facilitates accurate assessment of equipment fatigue damage.

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