SPECTRAL ANALYSIS OF VIBRATIONS EXPERIENCED BY PASSENGER OF RAILWAY VEHICLE IN SELECTED FREQUENCY BANDS

Received: December 2015

Summary: The paper presents the spectral analysis of the vibrations of a railway vehicle which a passenger is exposed to. The investigations have been performed for selected one-third octave bands with the centre frequencies from the range of 1.25 Hz to 20 Hz, which are related with different influence of vibrations on individual parts of a human body and their resonance frequencies. The simulation of the railway vehicle motion have been performed for various ride velocities. The obtained dynamical responses – the accelerations at selected points of the vehicle body are used for the spectral analysis of the vibrations experienced by a passenger.

Keywords: railway vehicle, passenger, vibrations, spectral analysis

1. INTRODUCTION

A passenger travelling by means of transport, especially in a railway vehicle, is exposed to vehicle vibrations, which lead to both whole-body and localized vibrations. This phenomenon is undesirable, since the action of mechanical vibrations can be uncomfortable or even harmful to human body and thus it should be reduced or eliminated. The negative effects of the exposition to the whole-body vibrations refer to the skeleton and the internal organs. On the other hand, mechanical vibrations transmitted to the human body through upper limbs (localized vibrations) lead to lesions in the cardiovascular, nervous and osteo-articular systems. Disturbances in functioning of internal organs due to the whole-body vibration result mainly from exciting the resonance vibrations of particular organs. It should be stressed that the “normal” modes of individual organs are not unambiguously defined since they depend on an adopted division of the human body into separate masses and the elastic and damping properties of the connections between these masses. These properties are dependent on the age, sex and dimensions of a person, etc. The frequencies of the “normal” modes of individual body parts also depend on the posture and the performed activities [8]. In the biomechanical model of a human body elaborated by R. R. Coerman [1] the resonance frequencies in the range below 20 Hz are mainly related to internal organs and skeleton. For instance, the abdominal organs correspond
to 4-8 Hz, the shoulder to 4-5 Hz, the arms to 5-10 Hz, and the spine to 10-12 Hz while the experimentally determined resonance frequencies of the chest organs are within the interval of 5-10 Hz [13]. In view of the differences among the resonance frequencies of different organs the effect of the external vibrations on the human body largely depends on the frequency of such vibrations.

The basic quantity taken into account in the assessment of the effect of the mechanical vibrations on the human body are the accelerations to which the person (passenger) is exposed to. In the quantitative evaluation of the vibration effect on the human body, its dependence on the vibration frequency is taken into account by determining the root-mean-square (rms) values of the accelerations in one-third-octave bands with different centre frequencies. Such spectral decomposition of the total acceleration into the frequency-specific rms accelerations, done separately for lateral and vertical directions, is used in the ISO 2631-1 standard [4] published in 1985 and the corresponding norm PN-91/S-04100 [10] where such rms accelerations are compared with the reduced comfort boundary, the fatigue-decreased proficiency boundary, and the exposure limit, specified separately for each frequency band. This method has been applied to investigate the ride comfort in railway vehicles, e.g., in the previous works by the present author [6, 7, 8]. The rms accelerations in one-third-octave bands are also applied in the later version of the ISO 2631-1 standard (released in 1997) to determine the level of vibrational comfort with the weighted rms accelerations. The hazard of whole-body vibration and different methods for its investigation are discussed in the work [2]. The notion of comfort also includes other elements associated with environmental factors which influence people. According to the idea proposed by Griffin [3] it is very desirable to establish a synthetic indicators which depend on more than physical factors, including vibration, temperature and others.

The present paper is focussed on the vibrations which affect a passenger in the railway vehicle. The accelerations which a passenger is exposed to are determined as the accelerations of the vehicle body at different locations within its space which correspond to different positions of the passenger within the car body. The accelerations of the vehicle body as functions of time are found as the dynamical responses with the simulations of the railway vehicle motion. They are subsequently used to find their power spectral densities which are finally utilised to calculate the corresponding rms accelerations in the relevant one-third-octave bands. The obtained rms accelerations corresponding to different frequencies are further investigated to examine how they change with the ride velocity.

2. RAILWAY VEHICLE MODEL AND CALCULATION METHOD

The motion of a passenger car is simulated using a non-linear railway vehicle model of 27 degrees of freedom and linear characteristics of primary and secondary suspensions [6]. The model parameters, including stiffness constants and damping coefficients as well as masses and inertia moments of the vehicle body, bogies and wheelsets are the same as previously used in [6]. The non-linear wheel (UIC 60) and rail (ORE S1002) profiles are
Spectral analysis of vibrations experienced by passenger of railway vehicle in … 43

used and the forces at the wheel/rail contact are calculated with Kalker’s simplified nonlinear theory [5]. The vehicle moves with constant velocity \( v \) along a stiff, tangent track with geometrical irregularities: lateral and vertical deviations of the track centre line, variable track gauge and local superelevation. These irregularities are functions of the position \( x \) along the track and thus they play the role of kinematic excitations in the dynamics of the railway vehicle. Simulations of the vehicle motion at various ride velocities were performed for a 4000 m long section of the QN 2 class track (the track classes are defined according to the UIC 518 code [11]). The simulations used track irregularities measured experimentally and described in the work [6]. As a result various dynamical responses that describe the state of the vehicle are found. In particular the vehicle body accelerations \( a_y(t) \) and \( a_z(t) \) in the lateral (\( y \)) and vertical (\( z \)) directions are obtained as functions of time \( t \) in the numerical simulations.

The rms accelerations \( a_{y,\text{rms}}(f_k) \), \( a_{z,\text{rms}}(f_k) \) in the one-third-octave bands \( (f_k - \Delta f_k / 2, f_k + \Delta f_k / 2) \), with the centre frequencies \( f_k \) and the width \( \Delta f_k = 0.231 f_k \), can be calculated numerically with the aid of the following expression

\[
a_{\eta,\text{rms}}(f) = \left( \int_{f - \Delta f/2}^{f + \Delta f/2} S_{\eta\eta}(f') df' \right)^{1/2} \quad (\eta = y, z)
\]

Therefore, the calculation of the rms accelerations requires the determination of the power spectral density (PSD) \( S_{\eta\eta}(f') \) which is the function of the frequency \( f' \) and it is integrated over \( f' \) in the above formula.

The PSD is found from the vehicle body accelerations \( a_y(t) \) and \( a_z(t) \) obtained in the simulations, using the modified periodogram method by Welch [12]. In this method, for a specified frequency \( f \), the Fourier transform of \( a_\eta(t) \) is first calculated for each of \( M \) subintervals \( (T_0, T_k) \) of the same length \( \Delta T = T_k - T_0 \). The intervals cover the whole time interval \((0, T)\) corresponding the vehicle motion duration and they partly overlap in order to increase the method’s accuracy. Thus, the quantities

\[
S^{(m)}_{\eta\eta}(f) = \frac{2}{\Delta T} \left| \int_{T_0}^{T_k} a_\eta(t) e^{-j2\pi f't} dt \right|^2
\]

\((m = 1, \ldots, M)\) are obtained and the estimator of the PSD of \( a_\eta(t) \) is found as their arithmetic mean

\[
S_{\eta\eta}(f) = \frac{1}{M} \sum_{m=1}^{M} S^{(m)}_{\eta\eta}(f) .
\]
3. RESULTS - RMS VEHICLE BODY ACCELERATION

The accelerations have been determined at the two different positions within the vehicle body: the centre of mass \( r_{\text{COM}} \) and the point \( r_{\text{COM}} + (8.75\text{m},0,0) \) located in the front end of the body. Almost identical results were obtained for the rms accelerations at the symmetrically located point \( r_{\text{COM}} - (8.75\text{m},0,0) \) in the back end of the vehicle body. Therefore, the results are presented only for the centre of mass and the specified point in the front end of the body.

Fig 1. Rms lateral accelerations \( a_{y,\text{rms}} \) at the centre of mass of the vehicle body in the 1/3-octave bands with the centre frequencies: from 1.25 Hz to 4 Hz (upper panel) and from 4 Hz to 12.5 Hz (lower panel), for ride velocities in the range 60-140 km/h.
The rms lateral accelerations $a_{y,rms}(f_k)$ at the two selected points of the vehicle body are shown in the figures 1 and 2. These accelerations strongly grow with the ride velocity $v$ in the whole investigated velocity range of 60-140 km/h for the one-third octave bands with the centre frequencies $f_k$ larger than 2 Hz. The increase of $a_{y,rms}(f_k)$ with the ride velocity depends the frequency $f_k$. For $v > 100$ km/h the acceleration $a_{y,rms}(f_k)$ is largest at $f_k = 4$ Hz, and it decreases monotonically with decreasing $f_k$ within the frequency range 2-4 Hz and with increasing $f_k$ for the frequencies 4-20 Hz. The lateral acceleration $a_{y,rms}(f_k)$ is roughly twice larger at the position in the front of the vehicle body than at its centre of mass. However, the effect of the ride velocity $v$ on $a_{y,rms}(f_k)$ is very similar at both points for all investigated frequency bands.
Fig 3. Rms vertical accelerations $a_{z_{rms}}$ at the centre of mass of the vehicle body in
the 1/3-octave bands with the centre frequencies: from 1.25 Hz to 4 Hz (upper panel) and
from 4 Hz to 12.5 Hz (lower panel), for ride velocities in the range 60-140 km/h.

The dependence of the rms vertical acceleration $a_{z_{rms}}(f_k)$ on $v$ is more complex. It
roughly grows with the ride velocity in the most of the considered frequency bands,
especially for $v > 100\text{ km/h}$. However, the dependence of $a_{z_{rms}}(f_k)$ on $v$ becomes
strongly nonmonotonic for the frequencies 6.3Hz and 8Hz for which this acceleration
attains very similar values for $v = 60\text{ km/h}$ and $v = 140\text{ km/h}$; it also has a maximum at
100 km/h for $f_k = 8\text{ Hz}$. As for $a_{z_{rms}}(f_k)$, the vertical acceleration $a_{z_{rms}}(f_k)$ is almost
twice larger at the position in the front of the body than at the centre of mass but again,
apart from this scaling, it depends on $v$ the in a very similar way at both points.
Fig 4. Rms lateral accelerations $a_{z, \text{rms}}$ in the front end of the vehicle body in the 1/3-octave bands with the centre frequencies: from 1.25 Hz to 4 Hz (upper panel) and from 4 Hz to 12.5 Hz (lower panel), for ride velocities in the range 60-140 km/h.

4. CONCLUSIONS

The rms accelerations at the two selected points of the railway vehicle body, grow with the ride velocity in the most of the selected frequency bands related to the hazard of the whole-body vibrations. However, this roughly monotonic trend is not found in all cases, in particular for the rms vertical acceleration in some frequency bands. The strong increase of the lateral rms acceleration with increasing $v$ is found to start at some threshold ride velocities $v$ which are smaller for lower frequencies $f_k$ in the range $f_k \geq 4\, \text{Hz}$. This effect can be associated with a characteristic wavelength of track irregularities or the wheelset...
hunting whose frequency depends linearly on $v$. The achieved conclusion that the band frequency has strong effect on how the rms vehicle body accelerations depend on the ride velocity can be useful in investigations related to the different vulnerability of various human body parts to vibrations with specific frequencies.

**Bibliography**


**ANALIZA WIDMOWA DRGAŃ ODCZUWANYCH PRZEZ PASAŻERA POJAZDU SZYNOWEGO W WYBRANYCH PASMACZ CzęSTOTLIWOŚCI**

**Streszczenie:** W pracy jest przestawiona analiza widmowa drgań pojazdu szynowego, na które pasażer jest narażony podczas jazdy. Badania zostały przeprowadzone dla wybranych tercijowych pasm częstotliwości z zakresu 1.25 Hz do 20 Hz, związanych z różnym wpływem drgań na poszczególne części organizmu człowieka i ich częstotliwościami rezonansowymi. Symulacje numeryczne ruchu pojazdu zostały przeprowadzone dla różnych prędkości jazdy. Otrzymane odpowiedzi dynamiczne pojazdu szynowego – przyspieszenia wybranych punktów nadwozia pojazdu szynowego posłużyły do analizy widmowej drgań odczuwanych przez pasażera.

**Słowa kluczowe:** pojazd szynowy, pasażer, drgania, analiza widmowa