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EVALUATING METHODS OF WHOLE-BODY-VIBRATION EXPOSURE IN TRAINS

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ABSTRACT

This paper studies the whole-body vibration in trains which constitutes one aspect of the physical environment that can cause discomfort to passengers.
Modern methods of assessment use digital techniques. Accelerometers are usually mounted on the seat pan, the backrest and floor (although occasionally it is necessary to measure solely on the floor for standing passengers). Depending on the location, direction and standard to be used, a different method of signal processing and scaling is used for each accelerometer. Data are frequency weighted in order to model the human response to vibration in that location and direction. The most commonly used weightings are W_k, W_d, W_b and the Sperling filter, B(f).
The root-mean-square (r.m.s.) is the basis for most assessments of railway vibration. However, it is also possible to measure the vibration dose value (VDV) and the maximum transient vibration value (MTVV) in performing assessments according to ISO 2631-1. When undertaking assessment in accordance with ISO 2631-4 [11], a statistical method may also be used. Several criteria systems have been defined to assist users in interpreting results. These include tables of overlapping ranges of magnitudes (ISO 2631-1) and thresholds (e.g. Sperling’s method). According to these criteria, previous measurements of vibration in trains have established that it is not usually considered severe, but, at worst, “strong, irregular, but still tolerable” or “a little uncomfortable”.

KEYWORDS: whole-body-vibration, train, Sperling’s method

1. INTRODUCTION

A lot of factors influence passenger comfort in the rail transport systems, such as the environmental factors of noise level, visual stimuli, temperature and humidity. Other influential factors include the effect of vibration on task (such as reading, writing, eating or drinking) or the physical construction of the carriage or rail infrastructure, and there are also less tangible factors such as expectation (which could be biased by the price paid for the ticket or class of carriage) [6], [13], [17].

The two main standards used for whole-body vibration exposure assessment are BS 6841 (1987) [2] or ISO 2631-1 (1997) [10]; however, vibration experienced in railway vehicles is different from other forms of transport [3], and, therefore, numerous different analysis techniques are in common usage. The differences between vibration in a railway vehicle and other forms of transport can be summarised as follows:

- *Frequency content* In the frequency range of 0.5 to 2 Hz, the greatest resonance peaks are in the transverse and vertical directions. These vertical resonance frequencies are linked to the suspension characteristics of the railway vehicles. For example, for the TGV Duplex the low and high vertical resonant frequencies of the suspension system occur at 0.7 Hz and 6.7 Hz respectively [3]. For the transverse direction, the suspension system resonance is determined using only the secondary suspension, and for the TGV Duplex, the coupled pitch and rolling frequencies are 0.35 Hz and 1.4 Hz respectively [3]. Due to its length, the body of the railway carriage is not rigid, and a modern steel railway vehicle may have a resonant frequency of 8 Hz or above.

• *Low overall acceleration magnitudes* The acceleration magnitudes found on railways are relatively low. The example data presented in Section 2.6 shows a mean value of 22 measurements of ISO 2631-1 [10] weighted worst-axis seat-pan acceleration of 0.32 m/s². This is compared to typical ISO 2631-1 values of 1.28 m/s² for an off-road quarry truck, 0.56 m/s² for a bus, 0.50 m/s² for a lorry, 0.39 m/s² for a car, 0.85 m/s² for an armoured vehicle and 1.08 m/s² for a helicopter [20].

• *Statistical properties* Vibration measured on railway systems fluctuates due to factors such as the train speed, variations in track quality and passing over turnouts, and is therefore not statistically stationary. The presence of such variables has prompted the production of an additional standard, ISO 10056 (2001) [7], which includes statistical methods to account for the nature of the acceleration data.

2. EQUIPMENT

The accelerations on the x, y and z axis were measured with MAESTRO vibrometers and with 01dB triaxial seat-pad accelerometers, set up at the seat/driver separation surface [12]. Accelerometer locations are shown for measurement of whole-body vibration exposure in railway carriage seats as shown in Fig. 1.

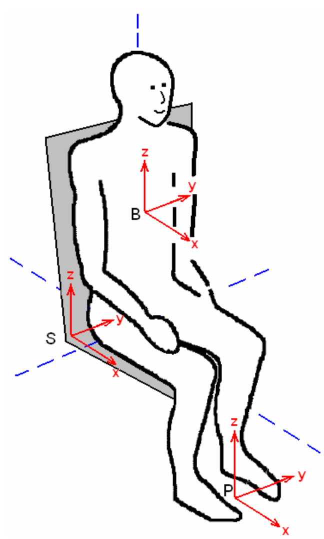

Fig. 1. Direction of vibration measurements and location of accelerometers for a railway vehicle, as defined in ISO 10326-2 (2001) [9]. 1 = Seat back, 2 = Seat pan, 3 = Floor (platform). (B) = Seat back accelerometer mounting position, (S) = Seat pan accelerometer mounting position, (P) = Platform accelerometer mounting position, ideally mounted directly under (S) and not to be mounted further than 100 mm off-centre (ISO 10326-1 (1992) [8]).

Accelerometer mounting positions are shown for the floor (P), the seat pan (S) and the backrest (B). The orientation of the three axes (x, y and z) is also shown for each mounting position.

3. DATA ANALYSIS

ISO 2631-1 Section 6 specifies an r.m.s. based method of evaluation of ride comfort. The weighted r.m.s. acceleration (in m/s²) of a discrete time-domain signal is given by:

$$a_{r.m.s.} = \sqrt{\frac{1}{N}\sum_{n=0}^{N-1} a_w(n)^2} \qquad (1)$$

where $a_w(n)$ is the n^{th} sample of the weighted acceleration, and N is the total number of samples in the measurement.

The crest factor is calculated to determine the most suitable method of analysis, and is given by:

$$CF = \left|\frac{\max(a_w)}{a_{w\,r.m.s.}}\right| \qquad (2)$$

If the crest factor is greater than 9, then the MTVV (maximum transient vibration value) or the VDV (vibration dose value) should be calculated in addition to the r.m.s.; however, it is also useful to present the MTVV or VDV for measurements with crest factors less than 9.

The MTVV takes into account occasional shocks and transient vibration by use of a short integration time. The MTVV is defined as the maximum value of the r.m.s. weighted acceleration calculated over a short time period (integration time) as given, in the discrete time-domain, by:

$$MTVV = \max\left(\sqrt{\frac{1}{\tau}\sum_{n=n_0}^{n_0+\tau} a_w\left(n^2\right)}\right), \qquad (3)$$

$n_0 = 0, 1, 2, \ldots, N-1-\tau$

where $a_w(n)$ is the current sample of the weighted acceleration, τ is the integration time and N is the total number of samples in the measurement. ISO 2631-1 recommends the use of 1 second as the integration time, i.e. in the discrete time-domain set $\tau = f_s$, where f_s is the sampling frequency.

The VDV method uses the fourth power of the vibration magnitude, which is more sensitive to shocks than using the square as in the r.m.s. calculation. The unit of VDV is $m/s^{1.75}$, and VDV is given by:

$$VDV = \sqrt[4]{\frac{1}{f_s}\sum_{n=0}^{N-1} a_w(n)^4} \quad (4)$$

where $a_w(n)$ is the current sample of the weighted acceleration, f_s is the sampling frequency and N is the total number of samples in the measurement.

The individual axes may be combined to give a total weighted r.m.s. acceleration value, given as:

$$a_v = \sqrt{\left(k_x^2 \cdot a_{wx}^2\right) + \left(k_y^2 \cdot a_{wy}^2\right) + \left(k_z^2 \cdot a_{wz}^2\right)} \quad (5)$$

where a_{wx} is the weighted r.m.s. vibration of the x-axis (similarly for y and z-axis) and k is the axis multiplier given in Table 1.

Table 1. Selection of axis multiplier and weighting filter as defined in ISO 2631-1(1997)

Position	Application	Measurement location	Axis	Axis multiplier
Seated	Health	Seat pan	X	1.4
			Y	1.4
			Z	1
	Comfort	Seat pan	X	1
			Y	1
			Z	1
		Seat back	X	0.8
			Y	0.5
			Z	0.4
		Floor	X	0.25
			Y	0.25
			Z	0.4
	Perception	Seat pan	X	1
			Y	1
			Z	1
Standing	Comfort	Floor	X	1
			Y	1
			Z	1
	Perception	Floor	X	1
			Y	1
			Z	1
Recumbant	Comfort	Under pelvis	X	1
			Y	1
			Z	1
	Perception	Floor	$X_{note\ a}$	1
			$Y_{note\ b}$	1
			$Z_{note\ b}$	1

Notes: a: ISO 2631-1 uses term "vertical" rather than "x-axis"
b: ISO 2631-1 uses term "horizontal" rather than "y or z-axis"

For an analysis with respect to comfort, the total vibration magnitude is compared to the approximate indications of likely reactions to various magnitudes of overall vibration total values in public transport as stated in ISO 2631-1 and repeated here in Table 2.

Table 2. Approximate indications of likely reactions to various magnitudes of overall vibration total values in public transport as stated in ISO 2631-1.

Weighted vibration magnitude (total of three axes)	**Likely reaction in public transport**
Less than 0.315 m/s^2	Not uncomfortable
0.315 m/s^2 to 0.63 m/s^2	Little uncomfortable
0.5 m/s^2 to 1 m/s^2	Fairly uncomfortable
0.8 m/s^2 to 1.6 m/s^2	Uncomfortable
1.25 m/s^2 to 2.5 m/s^2	Very uncomfortable
Greater than 2 m/s^2	Extremely uncomfortable

An examination of Table 2 reveals that in both cases the acceleration at the seat pan is greater than that at the floor, indicating that the seat is actually amplifying the vibration. This relationship may be expressed numerically by the SEAT (Seat Effective Amplitude Transmissibility) value [19], which is given by:

$$SEAT\% = 100 \times \frac{r.m.s._{seat}}{r.m.s._{floor}} \quad (6)$$

or $$SEAT\% = 100 \times \frac{VDV_{seat}}{VDV_{floor}}$$

When the SEAT value is greater than 100%, the seat is amplifying the vibration, and when the value is below 100%, the seat is attenuating the vibration. Note that for each axis in the examples presented, the SEAT is not below 100%, i.e. the seat does not attenuate the vibration magnitude.

The EU Directive sets limits on both hand-arm and whole-body vibration in terms of risk, and does not cover passenger comfort [18]. The exposure limits are defined as an "action value" and a "limit value" and both r.m.s. values and VDV values are given.

The resulting r.m.s. values should then be compared with the limit and action values stated in the directive which are given in Table 3 below.

Table 3. Daily exposure limit and action values for whole-body vibration as specified in the EU Physical Agents (Vibration) Directive (European Commission -2002) [4].

Exposure Action Value (EAV)	Exposure Limit Value (ELV)
0.5 m/s^2 A(8) r.m.s.	1.15 m/s^2 A(8) r.m.s.
9.1 $m/s^{1.75}$ VDV	21 $m/s^{1.75}$ VDV

Once the vibration magnitude is obtained (for the most severe axis), it is possible to calculate the length of time that a vehicle may be operated before reaching the thresholds given in Table 6.

The time, in hours, to reach the Exposure Action and Limit Values using r.m.s. calculations is given by:

$$T_{EAV_{A(8)}} = \frac{8 \times 0.5^2}{a_w^2} \text{ and } T_{ELV_{A(8)}} = \frac{8 \times 1.15^2}{a_w^2} \quad (7)$$

where a_w is the weighted r.m.s. vibration magnitude at the worst-axis.

The time, in hours, to reach the Exposure Action and Limit Values using VDV calculations is given by:

$$T_{EAV_{VDV}} = t \cdot \left(\frac{9.1}{VDV}\right)^4 \text{ and } T_{ELV_{VDV}} = t \cdot \left(\frac{21}{VDV}\right)^4 \quad (8)$$

where t (h) is the measurement duration and VDV is the weighted VDV vibration magnitude at the worst-axis.

4. SPERLING'S RIDE COMFORT INDEX

In an attempt to provide a metric which correlates objective vibration measurement to the subjective parameter of ride comfort, Sperling's ride index has been proposed and is used by numerous research institutes and railway companies from around the world [1], [14], [16], [21], [22]. The ride comfort index is described by Garg and Dukkipati (1984) [5], and Kim et al. (2003) [15].

Sperling's ride index is defined as:

$$W_z = \sqrt[10]{\left(\sum_{i=1}^{n_f} W_{Z_i}^{10}\right)} \quad (9)$$

where n_f is the total number of discrete frequencies of the acceleration response of the railway vehicle identified by the FFT and z_i W is the comfort index corresponding to the i^{th} discrete frequency, given by:

$$W_{z_i} = \sqrt[10]{a_i^3 B\left(f_i^2\right)^3} \quad (10)$$

where a_i denotes the amplitude of the acceleration response (in m/s²) of the i^{th} frequency identified by the FFT and $B(f_i)$ a weighting factor, given by:

$$B(f) = k\sqrt{\frac{1.911f^2 + \left(0.25f^2\right)^2}{\left(1 - 0.277f^2\right)^2 + \left(1.563f - 0.0368f^3\right)^2}} \quad (11)$$

where k = 0.737 for horizontal vibration and 0.588 for vertical vibration.

The calculated ride index value is mapped onto a subjective scale, as shown in Table 4.

Table 4. Mapping from numerical score calculated using Sperling's method (W_z), with a subjective scale of ride comfort.

W_z	Subjective ride comfort
1	Just noticeable
2	Clearly noticeable
2.5	More pronounced but not unpleasant
3	Strong, irregular but still tolerable
3.25	Very irregular
3.5	Extremely irregular, unpleasant, annoying, prolonged exposure intolerable
4	Extremely unpleasant, prolonged exposure harmful

5. RESULTS AND DISCUSSION

The mean values of the results presented in their paper are also shown in Fig. 2.

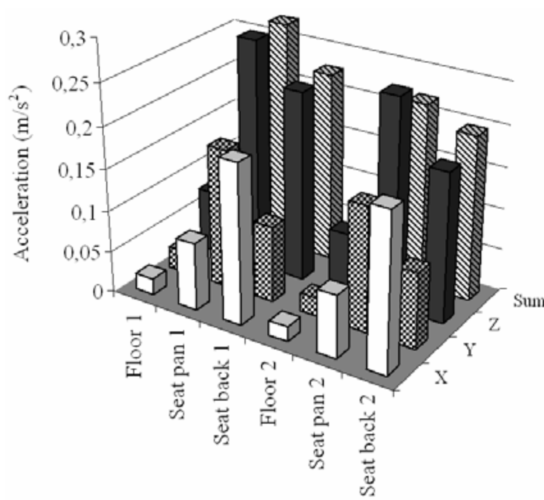

Fig. 2 Variation of acceleration on the x-, y- and z-axis as a function of accelerometers position

Fig. 2 shows that the vibration acceleration value is higher near the seat and these vibrations are transmitted along mainly the vertical direction. This is why the back pain appears often. Also it can be seen that the highest values of the acceleration were found for the z-axis, up to 0.275m/s^2.

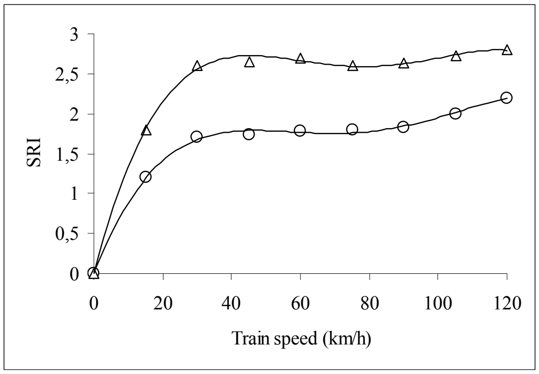

Fig. 3. Sperling's ride index for railway vehicles travelling at various speeds

Fig. 3 represents the Sperling's ride index (SRI) in two cases: a cargo train (Δ) and a passenger train (o). The Sperling's ride index value is quite high, up to 2.8 for the cargo train and this means that the subjective ride comfort is found between "*More pronounced but not unpleasant*" and "*Strong, irregular but still tolerable*". For the passenger train, the Sperling's ride index maximum value is 2.2 and this means that the subjective ride comfort is found between "*Clearly noticeable*" and "*More pronounced but not unpleasant*".

The curves of these variations are given by the equations 12 and 13:

(o) $SRI = -7\cdot10^{-8}\cdot v^4 + 2\cdot10^{-5}\cdot v^3 - 0{,}0024\cdot v^2 + 0{,}1099\cdot v + 0{,}0076$ (with: $R^2 = 0{,}9987$) (12)

(Δ) $SRI = -1\cdot10^{-7}\cdot v^4 + 3\cdot10^{-5}\cdot v^3 - 0{,}0036\cdot v^2 + 0{,}1676\cdot v + 0{,}0026$ (with: $R^2 = 0{,}9985$) (13)

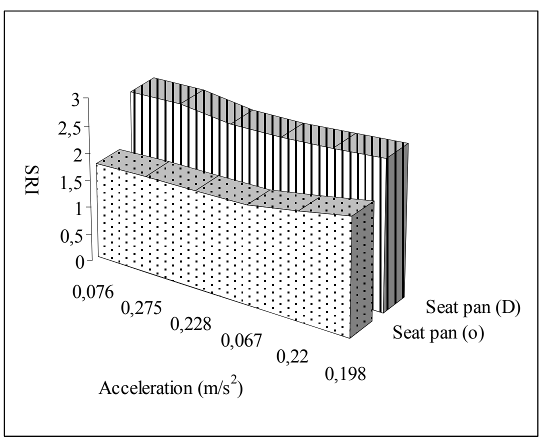

Fig. 4. Sperling's ride index for railway vehicles travelling at various accelerations (o) passengers train; (Δ) cargo train

After the data extrapolation, it can be said that the passengers train Sperling's ride index is de 1.93 and the cargo train index is 2.62, so it is placed in a value range which does not cause great discomfort to the passengers or to the working personnel.

Figure 4 shows the Sperling's ride index in terms of vibrations acceleration measured on passengers close to the seat and on the working personnel. It can be seen that the SRI does not depend on the value of this acceleration. The SRI remains almost constant around the 1.9 value for the passengers train and 2.68 for the cargo train.

6. CONCLUSIONS

There are several standardized methods of measurement and assessment of whole-body vibration in moving trains. This paper discussed the fundamental principles of these methods including a description of the measurement hardware and the necessary calculations that need to be carried out in order to comply with the relevant international standards. A number of different analysis methods have been discussed, all of which involve capturing the acceleration at the seat (and/or floor) of the train and processing to model the human response to the acceleration. Some of the techniques include methods to calculate expected passenger's comfort from the vibration magnitudes measured. Most techniques give results that indicate that the vibration in trains is not severe, but could occasionally cause some discomfort.

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