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Prediction of vehicle discomfort from transient vibrations

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Abstract

Vehicle manufacturers are continuously seeking to improve vibration comfort. In this paper, subjective responses from transient vibrations in a forklift were analyzed on the basis of ISO 2631-1 and a number of additional variables. The objectives were to define: the effect of different operating conditions and appropriate background variables of subjects on perceived motions; the development of model that describes perceived discomfort as a function of measured vibrations; and important frequencies for prediction of vibration discomfort. The experiment was based on 12 different operating conditions defined by the variables: vehicle speed, obstacle height and load conditions. Eleven professional drivers participated and their responses of overall discomfort were defined by a vector sum of three perceived motions: shaking, for-aft and up-down motions. The evaluation method, maximum transient vibration value as defined in ISO 2631-1 was found to be adequate in predicting vibration discomfort during a four second transient vibration exposure. By analysis of narrow frequency band spectra of vibrations several explanations for the test results are discussed. The best results were obtained using a prediction model based on accelerations in $\frac{1}{3}$-octave bands of pitch vibrations.

1. Introduction

Whole-body vibrations are well described by Griffin [1] and measurement of whole-body vibrations and comfort prediction is outlined in ISO 2631-1 [2].

Modeling of the dynamic behavior of off-road vehicles as a means to identify design solutions that reduce whole body vibration and increase operator comfort is of increasing interest.

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Improvements of vehicle vibration comfort can be carried out using dynamic computer simulations; this approach does require the use of a model that includes variables associated with vibration comfort. Important variables, extrinsic or intrinsic (as defined in the *Handbook of Human Vibration* [1] on page 45, Table 3.1), can be found by studying human response to vibrations.

A number of studies have addressed the problem of finding an appropriate technical method for judging the effect of mechanical shocks on discomfort [3–5]. Typically, only one direction of vibration has been studied. Holmlund [6] showed by using absorbed power that human response in one direction due to single-axis excitation is not readily transferable to a multi-axis environment. The reason for this is that single-axis vibration can have input in other directions due to the biomechanics of the human body.

Studies by Paddan and Griffin [7] showed that fore-and-aft vibration of the backrest of a seat can cause appreciable body vibration. Corbridge [8] demonstrated that this vibration can be a dominant cause of discomfort.

The current trend in vibration research is to use multi-axis values. This may be seen in studies such as those by Hinz et al. [9] and Paddan and Griffin [10]. Human response to whole-body vibrations is also affected by local vibrations, i.e. steering wheel hand-arm vibrations [11] and feet on the floor [2]. A study by Lundström [12] suggested a resonance region between 80 and 200 Hz for the finger and the hand. Logically, it then follows that it is reasonable to speculate that humans can assess higher frequencies at local parts of the human body, i.e. the skin when perceiving rapid movements such as during transients or shocks. Due to these reasons, prediction of vibration discomfort from transient vibration exposure (e.g. in off-road vehicles) could be improved by including assessments of the influence of higher frequencies and this may require different frequency weighting than what is described in ISO 2631-1 [2].

This study focuses on subjective responses and transient vibration exposures in a forklift. The vibration data was measured at 3 points and totally in 8 vibration directions during operating conditions and evaluated using the methods described in ISO 2631-1 [2]. The aim of this study was to investigate as to whether present methods yield appropriate predictions of vibration discomfort. The relationship between design variables (i.e. operating conditions) and the factors of subjective responses to vibrations and vibration measurements is also important when designing for optimal comfort.

The first objective was to investigate effects caused by different experimental design variables on subjective response and vibration accelerations. For subjective response, the effect of appropriate background variables (e.g. age and experience) on participating subjects was also investigated. The second objective was to define a model that described perceived discomfort as a function of important vibrations (individual points and directions). A third objective was to identify those frequencies which appear to most accurately correlate with vibration discomfort.

2. Methods and procedures

2.1. Subjects

Eleven subjects (the sample size) were selected and the statistical power was calculated and interpolated as 0.94 as described *Design and Analysis: A Researcher's Handbook* [13, pp. 76–79 and 516–517]. The power is based on an effect size Φ_A (a ratio of treatment variances relative to

Table 1
Minimum, maximum, mean and standard deviation of subject characteristics

Variable	Min	Max	Mean	SD
Age (year)	27	59	43.3	8.6
Weight (kg)	68	110	84.2	12.7
Body length (cm)	169	188	178.7	6.0
Years with employer	0	18	10.0	6.1
Years as professional driver	0	16	5.6	5.7

error variance multiplied to sample size) calculated from a small sample of data in a study by Mansfield [5] (11 male subjects, $\Phi_A = 1.62$, $\alpha = 0.05$). It is recommended that a statistical power greater than 0.80 [13, p. 69] be used. The subjects were forklift drivers working at the same forklift company. All had different additional duties related to manufacturing and/or forklift development. Background variables were collected by a questionnaire. Table 1 summarizes background variables seen as relevant and appropriate.

Six subjects were defined as short (169–176.5 cm in body length) and five as tall (180–188 cm in body length).

2.2. Experimental design

The experiment included 12 different test runs (T) carried out in random order with each subject. The experiment was a factorial design using three variables: vehicle speed, obstacle height and load condition. The effects of one or more variables on a response are often measured by designed experiments, i.e. factorial design [14, part III]. The technique of factorial design is described in *Statistics for experimenters* [14, pp. 306–309]. The variables of obstacle height and load condition contained two levels and vehicle speed contained three levels. The levels of the experimental variables were Load: no = 0 kg, yes = 3000 kg; Obstacle: low = 30 mm, high = 50 mm; Speed: 7, 12, and 20 km/h. The experimental design matrix is shown in Table 2.

2.3. Apparatus

The vehicle used in this study was an industrial forklift *(KALMAR-DCD80®)* manufactured by Kalmar Industries AB (Ljungby, Sweden). This model is equipped with four pneumatic front tires and two pneumatic rear tires. The suspension between each corner of cabin and chassis consists of 10 mm thick rubber bushings. The forklift used in the evaluation was equipped with a BE-GE® seat (manufactured by BE-GE Förarmiljö AB, Sweden). This seat has been described as having a cross-linkage mechanism, equipped with a squab and having a vertical adjustable pneumatic suspension located under the seat pan. The height of seat back was 640 mm.

An 8-channel Sony® DAT recorder and a charge amplifier (Bruel & Kjaer® 5974) were used for recording all vibration measurements. The sampling rate of the acceleration data was 24 kHz.The acceleration signals were low-pass filtered with the cut-off frequency set at 256 Hz.Two three-axis piezo-electric accelerometers (B&K® 4321) and two single-axis piezo-electric accelerometers

Table 2
The experimental design matrix

T	Load (kg)	Obstacle (mm)	Speed (km/h)
1	0	30	7
2	3000	30	7
3	0	50	7
4	3000	50	7
5	0	30	12
6	3000	30	12
7	0	50	12
8	3000	50	12
9	0	30	20
10	3000	30	20
11	0	50	20
12	3000	50	20

T stands for test number.

(B&K® 4366) and (B&K® 4371) were used to measure accelerations in three degrees of freedom (dof) in the occupant/seat and seat/cabin interface. The vibration axes in this study are in accordance with commonly used definitions and terminology [1, p. 35, Table 2]. Vertical (*sZ*) and for-aft vibrations (*sX*) at the occupant/seat interface were measured by using a pliable rubber disc embedded with one of the two three-axis accelerometers and placed on the seat pad according to ISO 2631-1 specifications. The first single-axis accelerometer was placed on a rigid beam mounted on top of the seat backrest and was used to measure accelerations in the *x*-direction (*bX*). The pitch motion at the occupant/seat interface, seat pitch (*SP*), was calculated by subtracting the accelerations in the *x*-direction on the top of backrest from the acceleration in the *x*-direction on the seat and then dividing the result by the distance between the measurement locations (0.6 m). The second three-axis accelerometer was mounted on the floor and located beneath the rear edge of the seat's centreline. This accelerometer was used to measure accelerations in the *x*- and *z*-directions at floor level. The second single-axis accelerometer was used to measure the accelerations in the *z*-direction of the floor and it was mounted on the floor beneath the front edge and centreline of the seat. The calculation of floor pitch (*FP*) at the seat/cabin interface was determined by subtracting the *z*-axis accelerations beneath the front edge of the seat from the *z*-axis beneath the rear edge of seat and dividing the result by the distance in the *x*-direction between the measurement locations (0.35 m). The centre of rotation around the *y*-axis of the vehicle was further forward in the *x*-direction for the loaded condition (point 1 in Fig. 1) when compared to the unloaded condition (point 2 in Fig. 1). The arrangements for the measurement of all conditions were identical for all drivers. Lateral, yaw and roll motions during the experiment were treated as being minimal; the experiment was designed so that these were minimized. The DAT recorder was placed on the right side of the driver for ease of access and operation. Fixed screws under the accelerator limited the vehicle's speed.

For subjective judgment of overall discomfort a bi-polar rating scale (similar to the rating scale used by Fothergill [15]) from 1 to 5 (where 1 is very uncomfortable, 2 is uncomfortable, 3 is neither uncomfortable nor comfortable, 4 is comfortable and 5 is very comfortable) were used. For

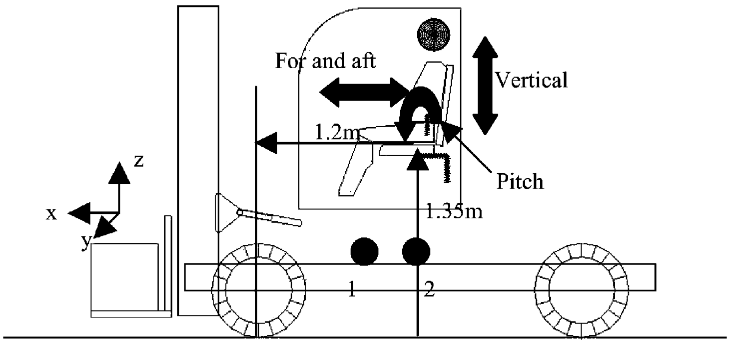

Fig. 1. The distance from the ground to the driver (1.35 m) and the distance from the front wheel to the driver (1.2 m). The distance between front and rear wheels is 2.4 m. Vibration in the x-direction is "for and aft", the z-direction is "vertical" and the pitch-rotation is in the seat; all are marked in the figure. Centres of rotation are marked by black dots; 1: loaded; and 2: unloaded.

perceived vibration motion three 9-cm continuous bi-polar scales were used. The scales were subdivided in 0.5 cm steps, 0–18 levels, correlating to three different perceived motions (shaking, for-aft and vertical). Assessments of for-aft and vertical were supposed to correspond to accelerations in the x- and z-directions, respectively. Seat pitch motions (*SP*) are commonly considered as a part of the assessments of for-aft motion due to the distance between the centre of rotation and the upper part of the back (see for example, Ref. [1]). The assessment of "shaking" was defined as a general perception of vibration commonly encountered during actual working conditions. The two end points for all three were "Very weak" and "Very strong". The three assessment scales of perceived motion used were based on descriptive word combinations familiar to the subjects. Different subjects might interpret differently a single scale, but if there are three different appropriate judgment scales corresponding to different vibration directions the correlation factor between measurement and discomfort is strengthened. The judgments were calculated as a vector sum of the judgments of shaking, vertical, and for-aft motions (Eq. (1)). This is the VSPM or Vector Sum of Perceived Motions. The VSPM consequently varies between 0 and 31.17 ($(18^2+18^2+18^2)^{1/2}=31.17$).

$$\mathrm{VSPM} = (\mathrm{shakings}^2 + \mathrm{verticalmotions}^2 + \text{for-aftmotions}^2)^{1/2}. \tag{1}$$

2.4. Experimental procedures

Subjects were exposed to 12 vibration stimuli and data collection was carried out in field conditions designed to simulate actual working conditions (i.e. drivers used a brand of forklift used in their actual work activities with test run conditions equal or comparable to actual work activities).

Data collection for each subject was completed in approximately 40 min and each test run lasted from 30 to 130 s. The experiment was performed on a clearly laid out 160 m smooth asphalt track. The test run track consisted of two obstacles with different heights placed next to each other. The obstacles were symmetric rigid steel triangles 2 m in length in the x-direction and 30 and 50 mm in height. Although each subject used the same vehicle, driver seat, speed, load and test run track, the acquired accelerations at the seat i.e. vibration stimuli were not constant (for example, subject weight caused variation). For all vibration stimuli (the 12 test runs (T1–T12)) the mean of the weighted acceleration in the z-direction varied from 0.7 to 2.1 m/s^2 root-mean-square (rms). Four weighted transient vibrations (i.e. the z-direction on the seat for one driver) are illustrated in Fig. 2a–d. This data corresponds to the different operating conditions of low and high speed, low and high obstacle, with and no load. Data for the low and high vibration stimulus condition may be found in Fig. 2a and d. The duration of the transients are less than 2 s with magnitudes varying between 2.5 and 6.0 m/s^2.

Before the start of data collection the drivers were provided with information concerning the experimental procedure. The weight adjustment of the seat damping range in the vertical direction was set to the middle position. The inclination of seat pan and backrest were set to approximately 10° and 110°, respectively (as measured in photos) and were fixed during all test runs. Drivers were instructed to adjust the seat only in the x- and z-directions according to personal preference. Except for these individual changes, the seat settings were the same for all subjects. No instructions regarding posture or body angles were given. From visual inspection of video recordings it was observed that all drivers leaned against the backrest during vibration exposures. No other body angles were measured. To minimize possible bias, subjects followed a predefined randomized order of trials. Prior to each test run, oral and written instruction was given. The instruction concerned which speed, obstacle and load would be used. The test subjects

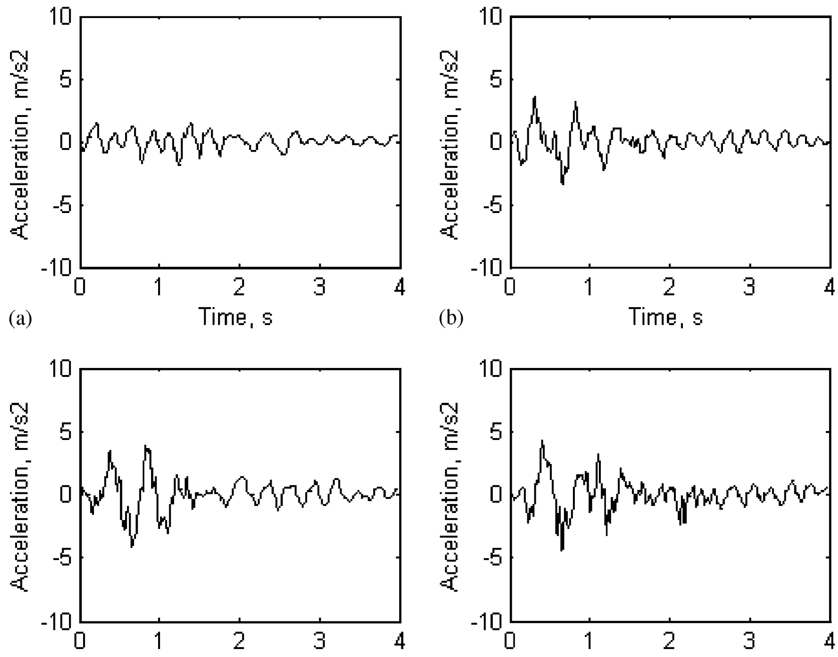

Fig. 2. Four different weighted acceleration signals in the z-axis on the seat are measured under different driving conditions. All time series are of the same driver (no.8). (a) Test3: 7 km/h, high obstacle and no load. (b) Test 9: 20 km/h, low obstacle and no load. (c) Test 12: 20 km/h, high obstacle and with load. (d) Test 11: 20 km/h, high obstacle and no load.

drove the forklift unloaded or loaded stable and straight in one direction over the obstacles at a constant speed. Two training trials were allowed at the start; conditions during the training trials were the same as those of the 11th and 12th actual trials (as noted earlier, these were randomly assigned). The reliability of the subjective response was checked by comparing the results of the training trials with the same actual test run ($r = 0.80$). Repetitions couldn't be performed since data collection was limited to two days. If using more time, the experimental conditions, e.g. weather conditions, might have been changed too much. Vibrations were measured throughout each test run and immediately after each test run drivers were asked to judge vibration discomfort. The general judgment question was: "How did you judge the shaking, vertical, and for–aft motions when you drove over the obstacle?" (Translated from Swedish). Each subject also made judgments on perceived over-all shaking, up–down motion, and for–aft motion.

2.5. Analysis

Analyses were performed using Matlab R12®, Mathworks® and I-deas®. The time history data for each driver and vibrations was cut into 4 s periods and weighted according to ISO 2631-1 [2]. According to ISO 2631-1 the manner in which vibration affects comfort is dependent on the vibration frequency content and is represented by the different frequency weightings. The weighting 'W_d' and 'W_k' was used for x-axis and z-axis motion measured at the seat pan, W_c was used for x-axis motion measured at the seat back, and 'W_e' applied to the seat rotational data at the occupant/seat interface.

When the crest factor is less than 9 the basic evaluation method is recommended by ISO 2631-1 [2], point 6.2.2. However, in this paper transient vibrations are studied and therefore analysis used both basic and additional methods.

The basic evaluation method for vibration is calculated using weighted rms acceleration defined by formula (1) from ISO 2631-1 [2] as shown below:

$$a_w = \left[\frac{1}{T}\int_0^T a_w^2(t)\,\mathrm{d}t\right]^{1/2}, \tag{2}$$

where T is the duration of measurement and a_w is the frequency weighted acceleration.

The additional methods used in ISO 2631-1 are vibration dose value (VDV) and maximum transient vibration value (MTVV).

The frequency weighted VDV is defined by formula (5) from ISO 2631-1 [2] as shown below:

$$VDV = \left\{\int_0^T [a_w(t)]^4\,\mathrm{d}t\right\}^{1/4}, \tag{3}$$

where T is 4 s.

MTVV is defined as the highest magnitude of $a_w(t_0)$ (index w stands for weighted data) according to formula (4) in ISO 2631-1 [2] as shown below:

$$\mathrm{MTVV} = \max[a_w(t_0)], \tag{4}$$

where $a_w(t_o)$ is defined by formula (2) from ISO 2631-1 [2] as shown below:

$$a_w(t_0) = \left\{ \frac{1}{\tau} \int_{t_0-\tau}^{t_0} [a_w(t)]^2 \, \mathrm{d}t \right\}^{1/2}, \tag{5}$$

where t_0 is instantaneous time, $a_w(t_0)$ is the instantaneous frequency weighted acceleration obtained by using the running rms evaluation method, integration time (τ) is 1 s.

The definition of vibration measurement is based on the additional method ([2, point 6.3]) indexes measurement direction and point, e.g. MTVV_{bX} (x-axis at seat back), MTVV_{sZ} (z-axis at seat), MTVV_{sX} (x-axis at seat), MTVV_{SP} (pitch rotational at seat). An overall vibration total value ([2, point 8.2.3]) is calculated and based on either MTVV or VDV defined as $\mathrm{MTVV}_{v\mathrm{sum}1}$ and $\mathrm{VDV}_{v\mathrm{sum}}$, respectively. The included measurement point and vibration axis in the overall vibration total value are; z-axis at seat pan (a_{wsz}), x-translation at the seat back (a_{wbx}) and pitch rotation of occupant/seat interface ($a_{w\mathrm{pitch}}$). The multiplying factors are in accordance with ISO 2631-1 [2] i.e. 'k_{sz}' = 1.0 m/s^2 (z-axis at seat pan), 'k_{bx}' = 0.8 m/s^2 (x-axis at seat back) and 'k_{pitch}' = 0.4 m/rad (applied to the seat rotational data).

The analysis procedure used for calculations is summarized as follows:

1: Import of field measurement data to analysis-software (Matlab®).
2: Calculation of rotational vibrations from vibration data in the x-directions of the seat and backrest.
3: Frequency weighting of vibration time history in each direction.
4: Calculation of MTVV and VDV and the vibration total value of $\mathrm{MTVV}_{v\mathrm{sum}1}$ and $\mathrm{VDV}_{v\mathrm{sum}}$.

A narrow frequency band spectra analysis of unweighted vibrations was performed to possibly explain the subjective response and vehicle behaviour. The data was unweighted as the floor vibrations beneath the seat are unweighted and were compared with the frequency response at seat interface. To avoid aliasing, the signals were low-pass filtered with a cut-off frequency of 100 Hz; which is above the highest frequency of interest. The sampling frequency was 256 Hz (following the Nyquist theorem).

For frequency analysis, the acceleration in the z-axis from the two floor accelerometers beneath the seat were averaged and calculated as "AFZ". The frequency analyses of AFZ, SZ and SP in four of the test runs (T7, T8, T11 and T12) were based on the "auto-power spectrum" (rms).

The $\frac{1}{3}$-octave spectra analysis was based on unweighted acceleration values (x-direction of the backrest, pitch and vertical motions on the seat) that recognized the correlation between subjective response and vibration frequencies from 1 to 100 Hz. The data was expressed in dB, centred and scaled to unit variance. The regression model used defines subjective response as a function of vibration levels in $\frac{1}{3}$-octave bands for different directions.

2.6. Statistical analysis

Analysis of variance of acceleration measurements and subjective response were carried out. The aim of these calculations was to study the main effects of each factor as well as the interaction between these factors. Analyses of Pearson product momentum correlations of subjective responses and acceleration measurements were performed. The aim of these calculations was to identify which factors might be used for the prediction of discomfort. Partial Least-Squares modeling (PLS) [16, p. 67; 17;18 p. 200], was used for multivariate analyses of the relationship between acceleration measurements, background variables and subjective responses. The software used for these statistical analyses were Windows Excel®, Statgraphics Plus 4.0® and Simca-P 7.01®.

3. Results

3.1. Analysis of MTVV and VDV in the seat driver interface

In Tables 3a–3c, results due to rms, MTVV and VDV analysis are presented as min, max, mean-values and standard deviations (SD) of *sX*, *sZ*, *bX* and *SP* for every test run.

Generally it can be seen that increasing speed and obstacle height gives higher vibration values. Highest vibration values are seen at speeds of 12 and 20 km/h for test runs without load and high obstacle. For example, in Table 3b, T7 ($sZ = 3.08\,\mathrm{m/s^2}$) and T11 (*bX*, *sX* and *SP* are 3.79, 1.81 $\mathrm{m/s^2}$ and $9.51\,\mathrm{rad/s^2}$, respectively). From Table 3b and c it can be seen that *sX* and *bX* are well correlated ($r = 0.99$). Vibrations in the x-direction on the seat (*sX*) are represented by vibrations *bX*, which is included in the seat pitch (*SP*) value. Corbridge [8] found that horizontal vibrations in a backrest (*bX*) might have a significant effect on comfort. Due to these reasons and to simplify analysis, the variable *sX* was excluded from further analysis.

3.2. Analysis and comparison of subject judgments

A first step in the analysis of subject responses was to establish a relationship between judgments of overall discomfort and perceived motions.

The three different judgments of perceived motions and VSPM are compared to the overall discomfort for each test run, Fig. 3. A regression model based on group averages for each test run, demonstrated an exponential and significant relationship between VSPM and overall discomfort (Fig. 3d, $r = 0.98$), Eq. (6), which also gave the highest correlation factor:

$$\mathrm{VSPM} = 0.77\,(\text{overall discomfort})^{2.43}. \tag{6}$$

Overall discomfort judgments of 2 corresponded to comfortable and 4 corresponded to uncomfortable. Accordingly VSPM = 4.1, corresponded to comfortable and VSPM = 22.4 corresponded to uncomfortable (Eq. 6).

Another advantage for the use of VSPM is that the correlation factor between perceived motion (VSPM) and $\mathrm{MTVV}_{v\mathrm{sum1}}$ ($r = 0.63$) was found to be higher than the correlation between overall discomfort and $\mathrm{MTVV}_{v\mathrm{sum1}}$ ($r = 0.57$).

Table 3

	bX (m/s^2)				*sX*				*sZ*				*SP* (rad/s^2)			
T	Mean	Min	Max	SD	Mean	Min	Max	SD	Mean	Min	Max	SD	Mean	Min	Max	SD
(a) Mean, minimum, maximum and standard deviation of rms																
1	0.5	0.4	0.6	0.1	0.3	0.2	0.4	0.1	1.1	0.7	1.3	0.2	0.7	0.5	0.8	0.1
2	0.6	0.4	0.8	0.1	0.4	0.2	0.7	0.1	0.7	0.5	1.1	0.2	0.9	0.5	1.1	0.2
3	0.6	0.5	0.7	0.1	0.4	0.3	0.6	0.1	1.2	1.0	1.4	0.1	0.9	0.8	1.0	0.1
4	1.0	0.8	1.2	0.1	0.6	0.5	0.9	0.2	0.8	0.7	1,I	0.1	1.4	1.3	1.6	0.1
5	0.8	0.3	1.0	0.2	0.5	0.1	0.8	0.2	1.8	1.1	2.3	0.4	1.1	0.5	1.3	0.3
6	0.7	0.5	1.0	0.1	0.4	0.3	0.6	0.1	1.5	1.0	2.0	0.3	1.1	0.9	1.3	0.1
7	1.0	0.9	1.1	0.1	0.6	0.4	1.1	0.2	2.1	1.3	2.7	0.4	1.4	1.3	1.5	0.1
8	1.0	0.8	1.1	0.1	0.6	0.5	1.0	0.2	1.6	1.2	1.9	0.2	1.6	1.4	1.8	0.2
9	1.5	1.1	1.8	0.2	0.8	0.6	1.4	0.3	1.6	1.2	2.0	0.3	2.0	1.7	2.3	0.2
10	1.5	1.3	2.0	0.2	0.9	0.7	1.4	0.3	1.4	1.1	1.9	0.2	2.1	1.9	2.5	0.2
11	2.7	2.4	3.1	0.2	1.5	1.0	2.9	0.7	1.7	1.2	2.2	0.3	3.4	3.1	3.8	0.2
12	1.6	1.4	2.1	0.2	1.0	0.7	1.8	0.4	1.6	1.4	2.0	0.2	2.4	2.1	3.1	0.3
(b) Mean, minimum, maximum and standard deviation of MTVV's																
1	0.7	0.5	0.8	0.1	0.3	0.3	0.4	0.0	1.5	1.0	1.8	0.4	0.9	0.7	1.0	0.1
2	0.8	0.7	1.0	0.1	0.4	0.4	0.5	0.1	0.9	0.6	1.3	0.2	1.2	0.9	1.4	0.2
3	0.9	0.7	1.1	0.2	0.4	0.4	0.5	0.0	1.6	1.4	2.2	0.4	1.3	1.0	1.5	0.1
4	1.4	1.1	1.7	0.2	0.7	0.6	0.7	0.1	1.1	0.8	1.3	0.3	1.9	1.7	2.2	0.2
5	1.3	0.5	1.6	0.3	0.6	0.2	0.8	0.1	2.6	1.4	3.5	0.7	1.6	0.7	1.9	0.4
6	1.0	0.7	1.5	0.2	0.5	0.4	0.7	0.1	2.1	1.5	2.7	0.5	1.5	1.3	1.9	0.2
7	1.5	1.3	1.7	0.1	0.8	0.6	0.9	0.1	3.1	2.0	4.0	0.7	2.1	2.0	2.4	0.1
8	1.4	1.1	1.6	0.2	0.8	0.6	1.0	0.1	2.3	1.8	2.8	0.5	2.3	2.0	2.6	0.2
9	2.2	1.7	2.6	0.2	1.0	0.8	1.2	0.1	2.3	1.9	3.0	0.6	2.9	2.4	3.4	0.3
10	2.3	1.8	3.1	0.4	1.0	0.9	1.3	0.1	2.2	1.7	2.9	0.5	3.1	2.6	3.9	0.4
11	3.8	3.7	4.6	0.9	1.8	1.7	2.0	0.1	2.6	1.8	3.4	0.7	5.2	4.8	5.7	0.3
12	2.7	2.2	3.2	0.3	1.2	1.0	1.6	0.2	2.7	1.9	3.3	0.6	3.7	3.4	4.5	0.3
(c) Mean, minimum, maximum and standard deviation of VDV's																
1	3.8	2.9	4.5	0.5	1.7	1.4	2.2	0.2	7.7	5.2	9.3	1.2	5.1	4.1	6.1	0.6
2	4.3	3.3	5.2	0.6	2.2	1.7	2.7	0.3	5.1	3.5	7.9	1.2	6.2	4.4	7.1	0.8
3	4.8	4.1	5.8	0.6	2.4	2.2	2.6	0.2	8.7	7.5	10.7	1.1	7.2	5.8	8.3	0.7
4	6.9	5.9	8.5	0.7	3.5	3.2	3.9	0.3	5.7	4.7	7.9	0.9	10.2	9.3	11.3	0.7
5	6.5	2.7	8.1	1.7	3.0	1.2	3.9	0.8	14.0	7.2	17.4	3.3	8.5	4.1	10.1	1.9
6	5.3	4.2	7.4	1.1	2.9	2.3	3.7	0.4	11.1	7.7	14.4	2.0	8.0	6.5	10.3	1.1
7	7.8	6.9	8.7	0.7	3.9	3.3	4.6	0.4	16.6	10.3	20.9	2.9	10.9	10.0	11.8	0.6
8	7.7	6.0	8.9	1.0	4.2	3.3	5.1	0.5	11.9	9.1	14.5	1.7	12.5	10.8	13.9	1.2
9	11.2	8.8	13.0	1.2	5.1	4.5	6.1	0.6	12.3	9.5	15.4	1.9	14.7	13.3	16.6	1.2
10	12.0	9.6	16.2	1.8	5.6	4.8	6.7	0.7	10.7	9.2	14.5	1.6	16.5	13.7	20.3	2.0
11	20.6	18.3	25.5	2.6	8.8	7.9	9.6	0.6	14.1	9.5	17.1	2.5	25.8	23.0	29.9	2.3
12	13.8	12.1	16.7	1.5	6.3	5.7	8.2	0.7	13.2	10.9	16.5	1.6	19.4	17.6	23.8	1.7

The terms *bX*, *sX* and *sZ* stand for vibrations in the x-axis at the seat back, the x-axis and z-axis at the seat pan. SP is the pitch in the occupant/seat interface. *T* stands for test number.

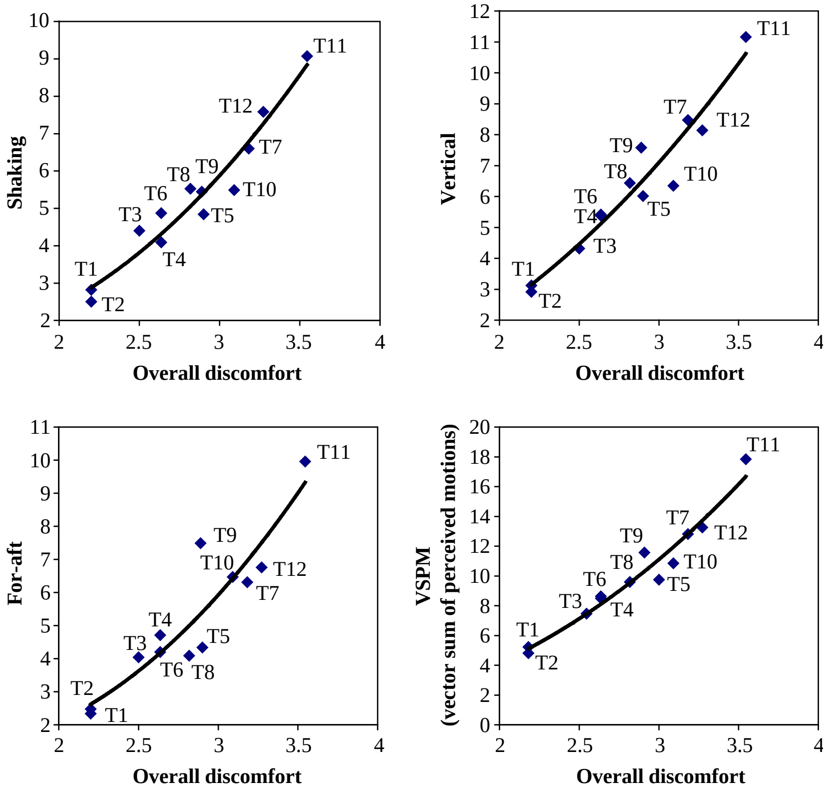

Fig. 3. The exponential relationship between subjective responses of perceived motions and overall discomfort for every test. The correlation factor 'r' for shaking, vertical, for–aft and VSPM was 0.97, 0.96, 0.92 and 0.98, respectively. The test numbers, Ti ($i = 1-12$), is according to the experimental design shown in Table 2.

3.3. Relationship between subjective response and vibration data

3.3.1. Difference between evaluation methods

The results of the evaluation methods are marked with different symbols for each subject (Figs. 4a–c) with similar patterns for all subjects. One driver did deviate significantly from the average, (marked by triangular-symbol, without fill).

Comparison of the methods shows apparent higher deviations in the case of VDV data (Fig. 4c) and smallest in the case of MTVV. Hence, the MTVV method was deemed to be a more accurate vibration discomfort indicator and was selected for use in further calculations.

3.3.2. Differences between individuals and the group

From normal probability plots, the subjective response and vibration data were found to have almost normal distributions (the kurtosis equal to 0.69 and -0.26 in case of VSPM and $MTVV_{vsum1}$, respectively), this means that median and mean values are similar. The observed variability on the measured vibration values among drivers (Fig. 4b) indicated a dependence on other variables. Points 1 and 2 (marked in Fig. 4b) illustrate an example of differences between

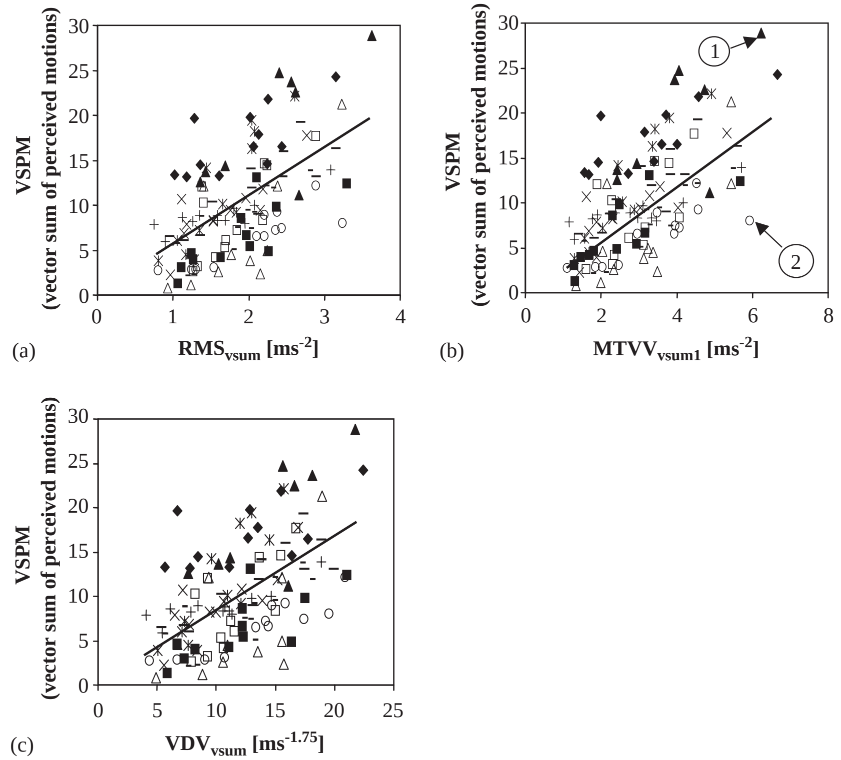

Fig. 4. Using $\mathrm{rms}_{v\mathrm{sum1}}$(a), $\mathrm{MTVV}_{v\mathrm{sum1}}$(b) or $\mathrm{VDV}_{v\mathrm{sum}}$(c) in predicting the subjective response (VSPM) gives different relationships between methods ($r = 0.61$,0.63 and 0.57, respectively) and subjects ($r = 0.62$–0.88). Eleven subjects are marked in figures; Subject 1 (◆), 2 (■), 3 (▲), 4 (×), 5 (⋇), 6 (○), 7 (+), 8 (–), 9 (—), 10 (△), and 11 (□). The black line indicates the average.

drivers during similar driving conditions. Points 1 and 2 correspond to no load, high obstacle and a 20 km/h speed for drivers 3 and 6, respectively. In Fig. 5, the data is based on group averages in each test run; this gave small deviations. Subject averaging always yields smaller deviations when compared to individual data. Individual differences such as experience, health, physiological body properties and different expectations are not averaged out.

VSPM is predicted from measured vibrations and by Eq. (1). In Table 4 the consistency of correlation coefficients between measured VSPM and predicted VSPM are summarized among the subjects. The judgments were consistent ($r = 0.79-0.82$) except in the case of subject 11 ($r = 0.62$).

The mean values and confidence intervals of $\mathrm{MTVV}_{v\mathrm{sum1}}$ and the judgments of perceived motions (VSPM) are shown in Figs. 6 and 7.

The spread in VSPM judgments are greater than the spread in acceleration measurement (Figs. 6 and 7). T11 (no load, high obstacle and 20 km/h) gave the highest mean-values of both VSPM and $\mathrm{MTVV}_{v\mathrm{sum1}}$. At the speed of 12 km/h, T7 (no load and high obstacle) gave the highest mean-values of both VSPM and $\mathrm{MTVV}_{v\mathrm{sum1}}$. Since VSPM and $\mathrm{MTVV}_{v\mathrm{sum1}}$ are dependent on the test variables (speed, obstacle and load), the following statistical analysis was performed to determine the effects from the experimental design variables.

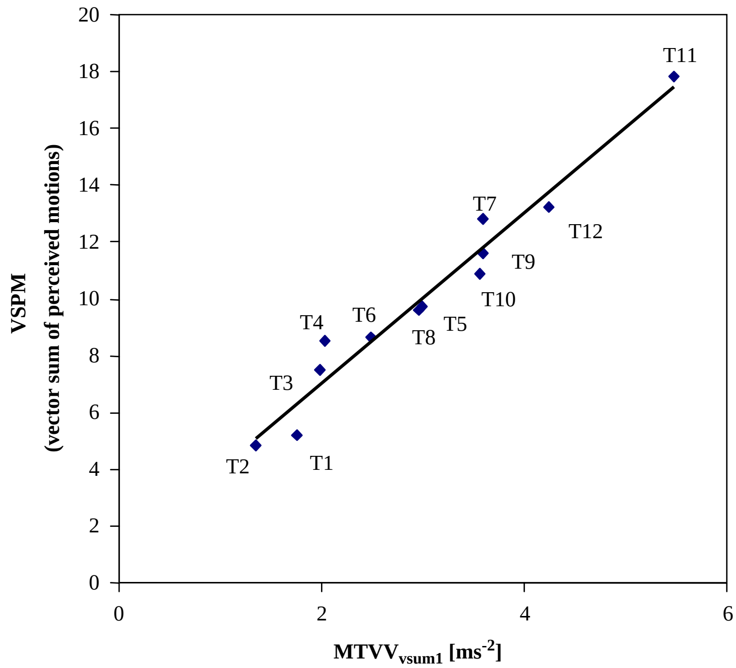

Fig. 5. Using $MTVV_{vsum1}$ in predicting the subjective response (VSPM), based on group averages in each test (T1-T12) ($r = 0.98$). Test numbers, Ti ($i = 1$–12), are shown (Table 2).

Table 4
Correlation coefficients between VSPM and predicted VSPM (average of all tests for each driver)

Driver	Average, VSPM	Correlation, r
1	17.10	0.79
2	6.47	0.79
3	18.94	0.80
4	8.93	0.82
5	11.74	0.87
6	6.42	0.81
7	8.88	0.88
8	7.94	0.81
9	11.85	0.84
11	8.80	0.62

3.4. The effect of experimental design variables

Main- and interaction effects were analysed with factorial design. In *Statistics for Experimenters* [14, pp. 317–318], it is stated that main effects can only be interpreted separately if there is no evidence that the variables interact with other variables. In case of VSPM (Table 5)

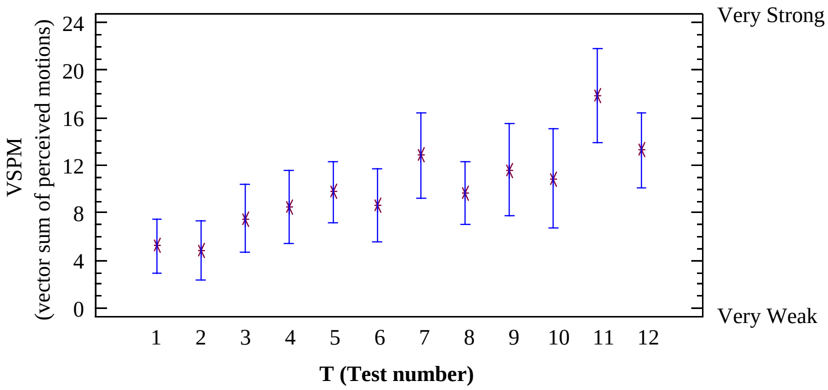

Fig. 6. Mean values and 95% confidence intervals ($N = 11$) of VSPM for T (test number) 1–12 (cf. Table 2).

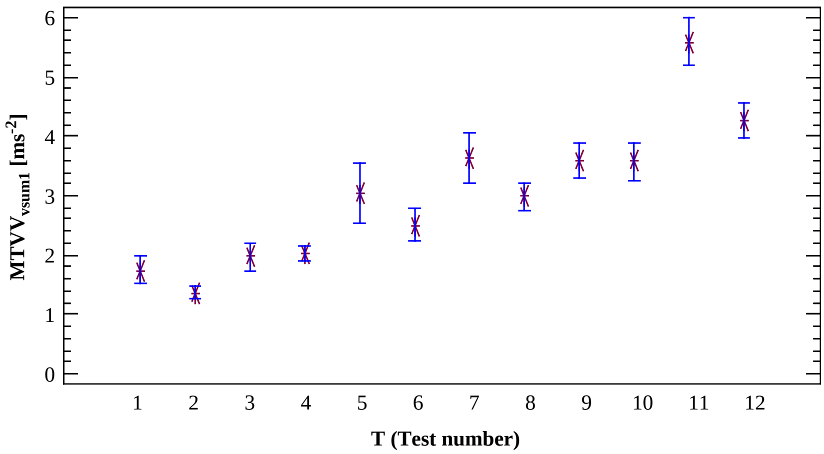

Fig. 7. Mean values and 95% confidence intervals ($N = 11$) of MTVV$_{vsum1}$ for T (test number) 1–12 (cf. Table 2).

two main effects were significant (speed and obstacle height) and there was no evidence of any interaction effects. The change in speed from 7 to 20 km/h increased VSPM from 6.6 to 13.5. The change in obstacle height from low to high increased VSPM from 8.5 to 11.6.

In case of MTVV$_{sZ}$, all three main effects were significant (Table 5) and there was no evidence of any interaction effects. The change in speed from 7 to 20 km/h increased MTVV$_{sZ}$ from 1.5 to 2.7 m/s^2. The change from low to high obstacle increased MTVV$_{sZ}$ from 1.9 to 2.2 m/s^2. The change from unloaded to loaded condition decreased MTVV$_{sZ}$ from 2.3 to 1.9 m/s^2.

In the case of MTVV$_{bX}$ and MTVV$_{SP}$, two interactions were significant (Table 5). MTVV$_{bX}$ and MTVV$_{SP}$ were highly correlated ($r = 0.99$), therefore interpretation focused on the results of MTVV$_{SP}$. The first interaction effect was between obstacle height and speed. With the low obstacle a change in speed from 7 to 20 km/h increased MTVV$_{SP}$ from 1.5 to 5.0 rad/s^2. With the high obstacle a change in speed from 7 to 20 km/h increased MTVV$_{SP}$ from 2.1 to 7.4 rad/s^2.

Table 5
P-values calculated for the main- and interaction effect of VSPM, $MTVV_{sZ}$, $MTVV_{bX}$ and $MTVV_{SP}$

Source	*P*-value			
	VSPM	$MTVV_{sZ}$	$MTVV_{bX}$	$MTVV_{SP}$
Load	0.0690	0.0003	0.1344	0.0744
Obstacle	0.0002	0.0131	0.0000	0.0000
Speed	0.0000	0.0000	0.0000	0.0000
Load∗obstacle	0.3602	0.7727	0.2138	0.0708
Load∗speed	0.1436	0.0545	0.0001	0.0000
Obstacle∗speed	0.4963	0.4743	4.0422	0.0001

The second significant interaction effect was between load and speed. During the condition "low speed", changing the load condition from "unloaded" to "loaded" increased $MTVV_{SP}$ from 1.4 to 2.3 rad/s^2. During the condition "high speed", increasing the load from "unloaded" to "loaded" decreased $MTVV_{SP}$ from 6.9 to 5.4 rad/s^2.

3.5. Frequency analysis of vibration signals

The excitation frequencies caused by the wheel impact are dependent on speed and wheel axis distance; this had a great effect on the measured vibrations. The fundamental excitation frequency at 12 km/h was 1.4 Hz, with harmonics at 2.8, 4.2, 5.6, and 7.0 Hz. At 20 km/h the fundamental frequency was 2.3 Hz, with harmonics at 4.6, and 6.9 Hz. Wheel impact excited two phenomena: the pitch and vertical motion of the vehicle and seat/driver interface. When the fundamental excitation frequency or its harmonics matched with pitch eigenfrequency, (the rear wheel is out of phase with pitch motion of the vehicle) the pitch motion diminished and only vertical motion remained.

The pitch motion was analysed by comparison of the frequency spectra during four test runs as shown in Fig. 8. *FP* stand for pitch vibration on the floor and *SP* stands for pitch vibration on the seat.

At 20 km/h, the frequency spectra (Fig. 8b and d) showed peaks at 2.5 Hz and 2.8 Hz, respectively. At 12 km/h in the unloaded condition (Fig. 8c) the peak cancelled. At 12 km/h and loaded condition (Fig. 8a), the amplitude at 2.5 Hz was reduced to about 25% of the amplitude value at 20 km/h (Fig. 8b). *SP* vibration below 4 Hz (peaks were at 2.5–2.8 and 3.5–3.8 Hz) was higher, for both T7 (Fig. 8c) and T11 (Fig. 8d), as compared to loaded configurations (Fig. 8a and 8b).

The vertical motion was analysed in a similar way to pitch motion (Fig. 9). AFZ stands for vertical vibration on floor and *sZ* stands for vertical vibration on the seat.

In Figs. 9a and c, clear and distinct peaks occurred at 3.5 Hz and 3.8 Hz, respectively. Peaks at 2.5 and 2.8 Hz due to pitch motion were diminished, which was in accordance with the observation that the excitation of the rear wheel was out of phase with the pitch motion of the whole vehicle.

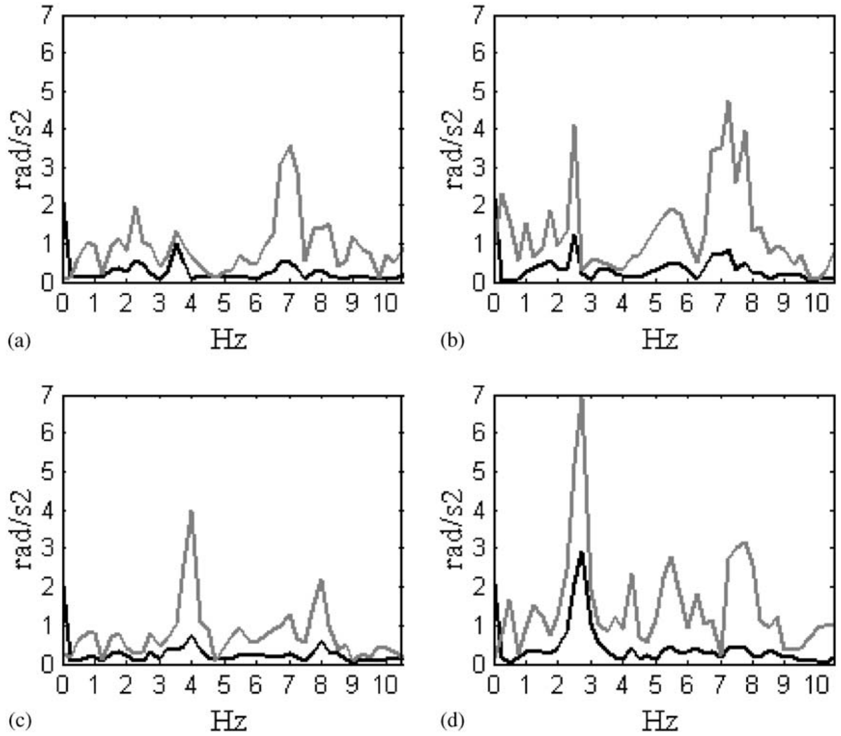

Fig. 8. Spectra of the pitch acceleration measured at the floor (*FP*—black line) and at the seat (*SP*–grey line) during the condition high obstacle with load (on the top) and without load (on the bottom) during a speed of 12 km/h (on the left) and 20 km/h (on the right). All spectra are for same driver (no.8).

Acceleration measured on seat surface was, in comparison to the acceleration on the floor (i.e. input), amplified within the frequency range 1–2 Hz (Fig. 9d). Consequently, frequencies well above this range were attenuated by the seat spring–damper system.

3.6. Background variables

In order to find a better model of subjective response than is given in ISO 2631-1 [2], a number of additional variables were tested. The background variables ‘‘body length’’ (Table 1) was found to have a significant effect on VSPM ($p = 0.0014$) and on $\mathrm{MTVV}_{v\mathrm{sum1}}$ ($p = 0.0291$). The variable ‘‘body length’’ slightly increased the correlation coefficient (from 0.63 to 0.65) between measured vibrations and vibration discomfort. Tall drivers perceived less VSPM than did short drivers (Fig. 10). The rank of VSPM is consistent.

3.7. Analysis based on $\frac{1}{3}$-octave band data

In order to analyse the relationship between perceived motions and measured vibrations as a function of frequency, an alternative prediction model was developed on the basis of PLS regression as described in Refs. [16, p. 67; [17,18], p. 200].

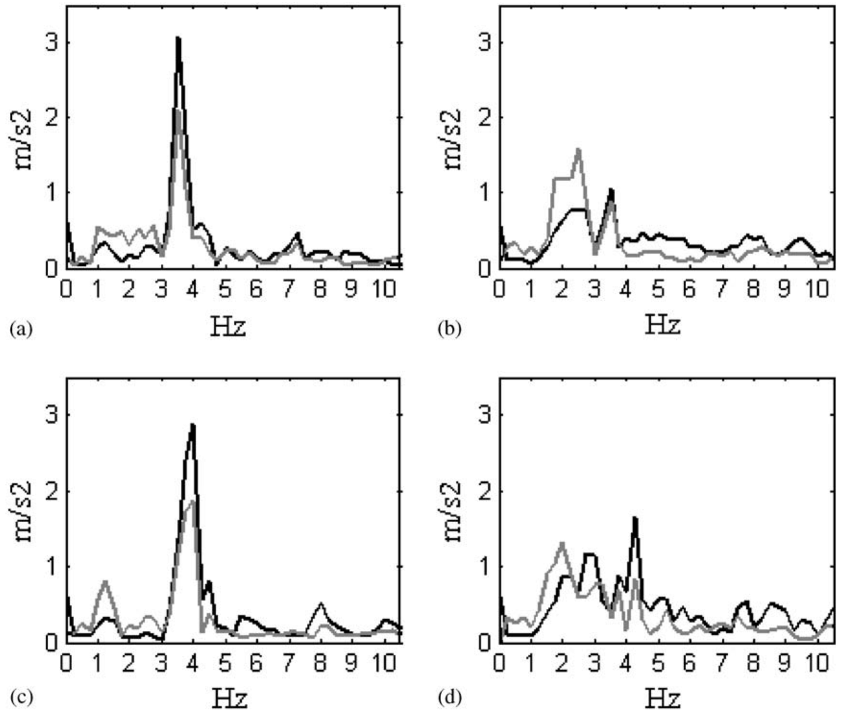

Fig. 9. Spectra of the z-axis acceleration measured at the floor (AFZ–black line) and at the seat (*sZ*–grey line) during the condition high obstacle with load (on the top) and without load (on the bottom) during a speed of 12 km/h (on the left) and 20 km/h (on the right). All spectra are for same driver (no. 8).

Pitch vibrations (*SP*) correlated significantly higher with subjective response than did *sZ*. The relationship between observed VSPM and predicted VSPM (based on *SP*) is shown in Fig. 11 ($r = 0.63$).

Analysis of $\frac{1}{3}$-octave band data showed the importance of the use of different frequencies in predicting discomfort. The results showed that the variability of frequencies above 50 Hz is of importance in predicting VSPM (Fig. 12). Frequencies of 1.6, 6.3 and 10 Hz were also of high importance. Frequencies from 2 up to 4 Hz were of low importance.

Predicted VSPM as a function of pitch vibrations in third octave bands were defined by a regression model:

$$\mathrm{VSPM} = \beta_0 + \sum_{i=1}^{20} \beta_i * X_i, \tag{7}$$

where i is the ith $\frac{1}{3}$-octave band.

The constant of proportionality β_0 was 0.95 and the 20 regression coefficients β_i are given in Fig. 12.

The prediction model provides a means to evaluate discomfort in relation to vibration spectra produced by the studied vehicle and road conditions. This model (Eq. (7)) could serve as a tool for

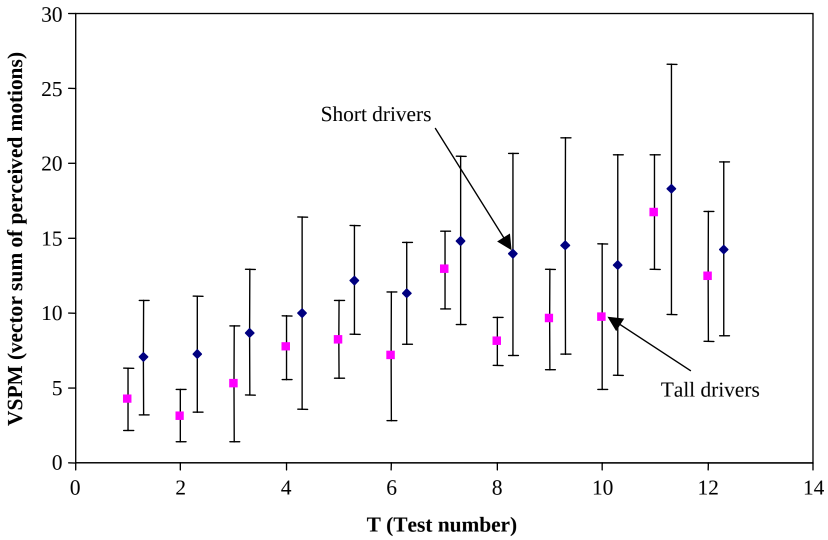

Fig. 10. VSPM for every test, based on short and tall drivers. The 'short' and 'tall' group consisted of 6 and 5 subjects, respectively.

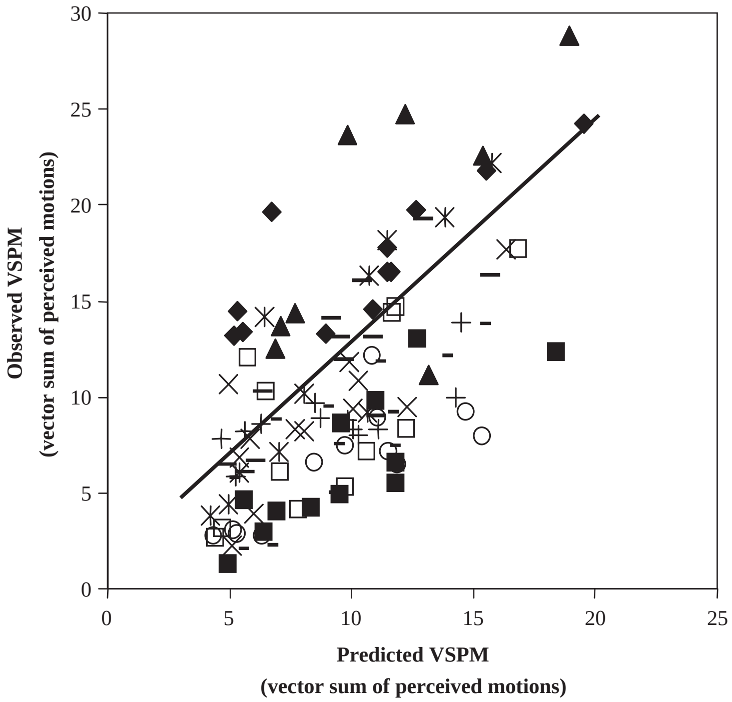

Fig. 11. Comparison of observed VSPM with the predicted VSPM ($r = 0.63$). Predicted VSPM is calculated from pitch vibration values in $\frac{1}{3}$-octave bands (Eq. (7)). Data is logarithmic. Ten subjects are marked in figures; Subject 1 (◆), 2 (■), 3 (▲), 4 (×), 5 (⋇), 6 (○), 7 (+), 8 (–), 9 (—), and 11 (□). The black line indicates the average of the drivers.

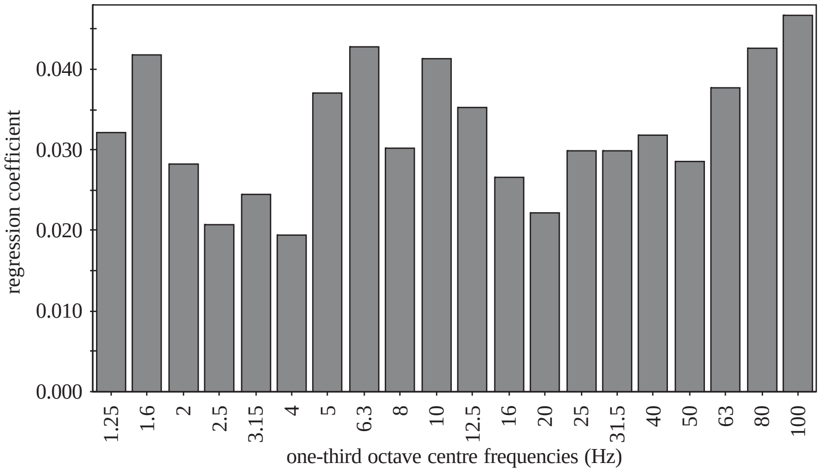

Fig. 12. The importance of the pitch frequencies on the vector sum of perceived motions (VSPM). Data is logarithmic.

the evaluation of changes in discomfort levels due to any modification of the studied vehicle. Attenuation through seat design at these pitch frequencies is an additional possibility that can be explored in the future.

4. Discussion and conclusions

Vibrations were measured in a forklift truck with goal of better understanding the relationship between subjective response to vibration transients and measured accelerations when designing for optimal comfort. The first steps were to decide which measurement—and analysis methods to use.

The method used to measure subjective response was based on a study by Fothergill [15]. The assessment question in this study (‘‘How did you judge the shaking, vertical, and for-aft motions when you drove over the obstacle?’’) asked the subject to judge a single transient in the vibration exposure in terms of overall discomfort and motions as shaking, vertical and for-aft. Discomfort was additionally defined by a vector sum of perceived shaking and motions in the vertical and horizontal directions, VSPM. Subjective responses to different test run conditions showed that speed and obstacle height had significant effects on VSPM. VSPM (perceived motions) was chosen as an indicator of discomfort as the correlation factor between VSPM and vibration measures (based on individual data) was greater, i.e. in case of $\text{MTVV}_{v\text{sum}1}$, the correlation factor increased from 0.57 (overall discomfort) to 0.63 (VSPM). The differences in VSPM (Fig. 10) for the two length groups (tall and short subjects) may be caused by one or two factors. First, increased body length may reduce those body vibrations that influence discomfort perception. Second, vibrations due to body biodynamics affect subjects with different body lengths differently. Significant design variables (first objective) can be used in simulation computation. It was found that all three design variables (speed, obstacle and load) were significant in the case of MTVV_{sZ}. In the case of MTVV_{SP} (and MTVV_{bX}), two interactions were significant (obstacle∗speed and load∗speed).

The second objective was to test the existing ISO prediction guidelines used to describe perceived discomfort as a function of measured vibrations and, if appropriate, refine those guidelines. A method based on the MTVV-method of pitch and vertical seat vibrations ($r = 0.63$) was found to be accurate; this is a variation upon the options allowed by the ISO guidelines. From the study by Mansfield [5], the VDV-method was found to be preferable. Using Mansfield, the median of individual correlation factors between subjective response and measurement in the case of VDV and MTVV were found to be 0.83 and 0.74 respectively. The median of corresponding correlation factors in this study (Table 5) were lower when using the VDV-method ($r = 0.75$) and higher when using the MTVV-method ($r = 0.81$) when compared to Mansfield's study. The 0.63 value mentioned above was based on calculations that used data from each subjects whereas the 0.75 and 0.81 values were derived using Mansfield's methodology. A possible explanation for the higher correlation factor in case of VDV (in Mansfield's study) and MTVV (in this study) might be related to the assessment question wording and vibration duration. The assessment question ("How did you judge the shaking, vertical, and for-aft motions when you drove over the obstacle?") could be a limitation when using the VDV-method. Some subjects may judge vibration exposure before and after the transient while others, more correctly, are judging the short transient in the actual vibration exposure. Due to this, the spread increases and the correlation factor decreases. The question in the study by Mansfield ("How severe did you judge the vibration?") is viewed by these authors as more a question that asked subjects to judge the vibration exposure during a longer 20 s period. Second, the analysis time of 4 s possibly favours the MTVV method since VDV didn't increase more significantly than did MTVV (Fig. 13). Contradictorily, in the Mansfield study ([5, Fig. 3]), VDV increased with a 20 s time duration, but the MTVV did not increase.

Due to differences between controlled laboratory and field environment experiments (more uncontrollable factors) it is expected that there will always be a higher correlation factor with laboratory experiments.

In the frequency analysis, the fundamental excitation frequency or its harmonics were found to diminish the pitch motion (only vertical motion remained) when matched with pitch frequency. This is a possibly explanation for the decreased MTVV_{SP} and second interaction effect in Section 3.4. Further, from the frequency analysis, it's reasonable to conclude that the peaks at 2.5 and 2.8 Hz were related to the pitch motion of the whole vehicle. The increased inertia of rotation and additional mass from the load explains the decrease in eigenfrequency of pitch vibrations from 2.8 to 2.5 Hz (Figs. 8b and d) and the vibration decreased in the z-direction from 3.8 to 3.5 Hz (Figs. 9a and c).

At loaded and highest speed (Fig. 9d), the eigenvibration frequency of 3.5 Hz (in the z-direction at floor level) was present but was not present when unloaded and at highest speed. The reason for this is probably due to there being a shorter distance between centre of rotation and the seat in the case of the unloaded condition. The ISO standard states that the perception of rotation is high in the range 0.2–4 Hz ([2, Fig. 3]). From this, it's reasonable to assume that the high VSPM and $\mathrm{MTVV}_{v\mathrm{sum}1}$ in T7 and T11 (Figs. 6 and 7) were driven by the pitch in the range of 0.2–4 Hz.

The third objective which was to identify those frequencies which appear to most accurately correlate with vibration discomfort was fulfilled by the $\frac{1}{3}$-octave band analysis. $\frac{1}{3}$-octave band

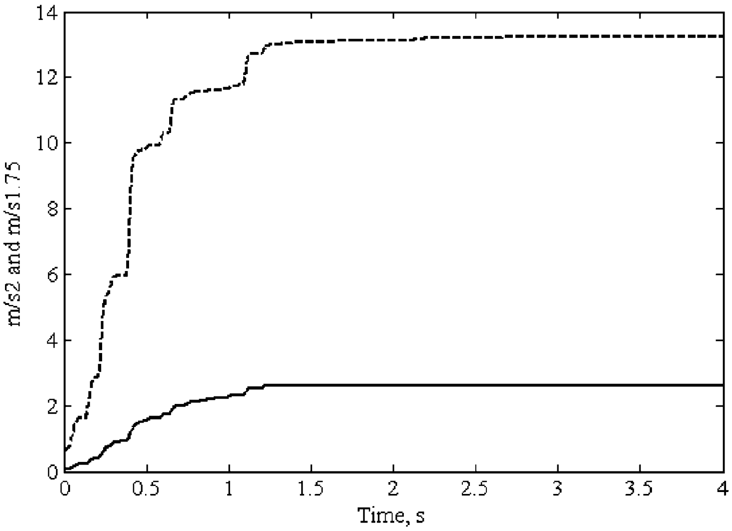

Fig. 13. Increasing MTVV (black line) and VDV (dashed line) during a single transient vibration of 4 s.

frequencies at 1.6, 6.3, 10 Hz and above 50 Hz were found to be most important in predicting vibration discomfort. These frequencies (especially the high frequencies) are related to the transient or rapid movements. In this study, these may have been caused by pitch or x-axis movements of the seat back since analysis of $MTVV_{sZ}$ showed high frequencies to be of a lesser importance. The seat wasn't provided with any pitch damping and therefore high frequencies were transmittable into the subject's bodies. Pitch frequencies from 2 up to 4 Hz were of less importance. This is somewhat unexpected as the eigenfrequencies of the vehicle are 3.5–3.8 Hz in the vertical direction and 2.5–2.8 Hz in the pitch direction. The explanation might be that these frequencies are natural in this particular vehicle and do not vary as much as the frequency bands that are related to forced movements, e.g. 1.6, 6.3 and 10 Hz.

In a multivariate analysis of $\frac{1}{3}$-octave spectra, backrest vibrations didn't improve prediction of vibration discomfort; therefore a regression model (Eq. (7)) based on pitch vibration in $\frac{1}{3}$-octave spectra was used. If using bX (i.e. when pitch can't be measured) a correct mathematical model of the back rest would be needed, and consequently, this would increase the complexity of the computer model.

Acknowledgments

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References

[1] M.J. Griffin, *Handbook of Human Vibration*, Academic Press, London, 1990.

[2] International Organization for Standardization, ISO 2631-1. Mechanical vibration and shock—evaluation of human exposure to whole body vibration–part 1: general requirements, 1997.

[3] B.-O. Wikstroem, A. Kjellberg, M. Dallner, Whole-body vibration: a comparison of different methods for the evaluation of mechanical shocks, *International Journal of Industrial Ergonomics* 7 (1991) 41–52.

[4] K. Spång, Assessment of whole-body vibration containing single event shocks, *Noise Control Engineering Journal* 45 (1997) 19–25.

[5] N.J. Mansfield, P. Holmlund, R. Lundström, Comparison of subjective responses to vibration and shock with standard analysis methods and absorbed power, *Journal of Sound and Vibration* 230 (2000) 477–491.

[6] P. Holmlund, Absorbed Power and Mechanical Impedance of the Seated Human Exposed to WBV, in Horizontal and Vertical Directions, Doctoral Thesis, Umeå University and National Institute of Working Life, 1998, ISBN 91-7191-457-9.

[7] G.S. Paddan, M.J. Griffin, The transmission of translational seat vibration to the head—II. Horizontal seat vibration, *Journal of Biomechanics* 21 (1988) 199–206.

[8] C. Corbridge, Predicting the discomfort of simulated vehicle rides, *Proceedings of the United Kingdom Informal Group Meeting on Human Response to Vibration*, National Institute of Agricultural Engineering, Silsoe, Bedfordshire, 14–16 September 1983.

[9] B. Hinz, H. Seidel, G. Menzel, R. Blüthner, Effects related to random whole-body vibration and posture on a suspended seat with and without backrest, *Journal of Sound and Vibration* 253 (2002) 265–282.

[10] G.S. Paddan, M.J. Griffin, Evaluation of whole-body vibration in vehicles, *Journal of Sound and Vibration* 253 (2002) 195–213.

[11] International Organization for Standardization, ISO 5349-1. Mechanical vibration—measurement and evaluation of human exposure to hand-transmitted vibration—part 1: general guidelines, 1999.

[12] R. Lundström, Local vibrations—mechanical impedance of the human hand's glabrous skin, *Journal of Biomechanics* 17 (1984) 137–144.

[13] G. Keppel, *Design and Analysis: A Researcher's Handbook*, Third ed, Prentice-Hall, Englewood Cliffs, NJ, 1991.

[14] G.E.P. Box, W.G. Hunter, J.S. Hunter, *Statistics for Experimenters: An Introduction to Design, Data Analysis and Model Building*, Wiley, New York, 1978.

[15] L.C. Fothergill, A Study of the Subjective Response to Whole-body Vibration, MSc Thesis, University of Southampton, 1972.

[16] L. Eriksson, E. Johansson, N. Kettaneh-Wold, S. Wold, *Introduction to Multi- and Megavariate Data Analysis using Projection Methods*, Umeå, Umetrics AB, 1999.

[17] A. Höskuldsson, *Prediction Methods in Science and Technology*, Thor Publishing, Copenhagen, 1996.

[18] M.S. Khan, Ö. Johansson, W. Lindberg, U. Sundbäck, Annoyance of idling diesel engine evaluated by multivariate analysis, *Noise Control Engineering Journal* 43 (1995) 197–207.