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Measurements and simulation on the comfort of forklifts

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Abstract

In order to determine the influence of some parameters of a forklift such as the road profile, the tyre characteristics, the riding comfort, etc., measurements carried out on a forklift with different tyres and seats were evaluated using different standards and methods. In addition, a simulation model was developed and used to investigate the influence of these parameters. Simulations and test run results showed good agreement.

The comparison of the results obtained with several methods of comfort evaluation and a series of tests showed that they nearly all resulted in the same classification. However, the results obtained with different methods could not always be compared among themselves.

Solid tyres were found to be more comfortable than pneumatic ones because of their high damping. The negative influence of higher stiffness was smaller than the positive influence of higher damping. The simulations pointed out that for a global general investigation about comfort, the influence of the horizontal tyre stiffness and damping can be neglected. Also the seat characteristics could be linearized. When the stability of the forklift has to be investigated, the horizontal forces must also be considered.

1. Introduction

For terrain and road vehicles it is clear that the riding comfort is better with pneumatic tyres than with solid ones. In the case of forklifts, however, there exists some doubt. During tests, in which the drivers do not know the type of tyres used, the drivers find the solid tyres more comfortable. At the moment, however, when they find out that the forklift is equipped with solid tyres, they change their mind. It means that there is a big psychological influence. In this study, it is investigated how to translate the comfort level into an objective value in order to exclude the subjective feeling of the driver. To this end, measurement results on a forklift as well as simulation

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results were used. Both measurements and simulations have inconveniences. Measurements are only short time recordings of signals that are very time variable. The biggest problem is the interpretation of the record. Simulations are always a simplification of the real vehicle.

The comfort of a forklift means here work proficiency and health influenced by mechanical vibrations. The relations between vibrations and these parameters, especially spinal load, are only speculative because of insufficient data in literature.

The global simulation of a forklift is very complicated. Therefore, a relatively simple model is considered, without a lot of local simulations. The goal of this study is to establish a tool which enables one to investigate the influence of parameters and to draw conclusions in a relatively simple, quick and inexpensive way.

2. Comfort evaluation

2.1. Generalities

Mechanical vibrations influence the comfort, the ability to work and even the health of the driver. During the ages several evaluations of these influences were proposed, but at this stage all the accepted methods rely on the weighting curves of the 2631 standard of the International Organisation for Standardisation (ISO) [1]. Sometimes the application and the elaboration of the results are different. More about this methods and their relationship can be found in Ref. [2]. Here a short description will be given of the methods used in this investigation.

2.2. Standards and methods

2.2.1. ISO 2631

Fig. 1 shows the normalized vibration exposure criterion curves for vertical (i.e., parallel to the z-axis) vibrations transmitted to the torso of a standing or sitting human being. The vibration levels indicated by the curves are given in terms of root mean square (r.m.s.) acceleration levels that produce equal decreasing proficiency due to fatigue. Exceeding the exposure time will—in most situations—cause noticeable fatigue and decrease job proficiency. These curves were originally drawn up for the case of pure harmonic vibrations. In the case of a broadband vibration, the r.m.s. values obtained in third octave bandwidths have to be calculated. These values are to be treated as pure harmonic vibrations and compared with the given curves. The highest weighted value is determining for the allowed exposure time. Note that the inverse of the indicated curves can be considered comfort weighting curves where the gain in frequency band between 4 and 8 Hz is taken equal to unity.

Analogous curves exist for the longitudinal (x-axis) and traversal (y-axis) directions.

Multidirectional vibrations can be estimated using the equation.

$$a = \sqrt{(1.4a_{xw})^2 + (1.4a_{yw})^2 + a_{zw}^2}, \tag{1}$$

where a_{iw} is the acceleration in direction i and weighted according to the concerning ISO curve.

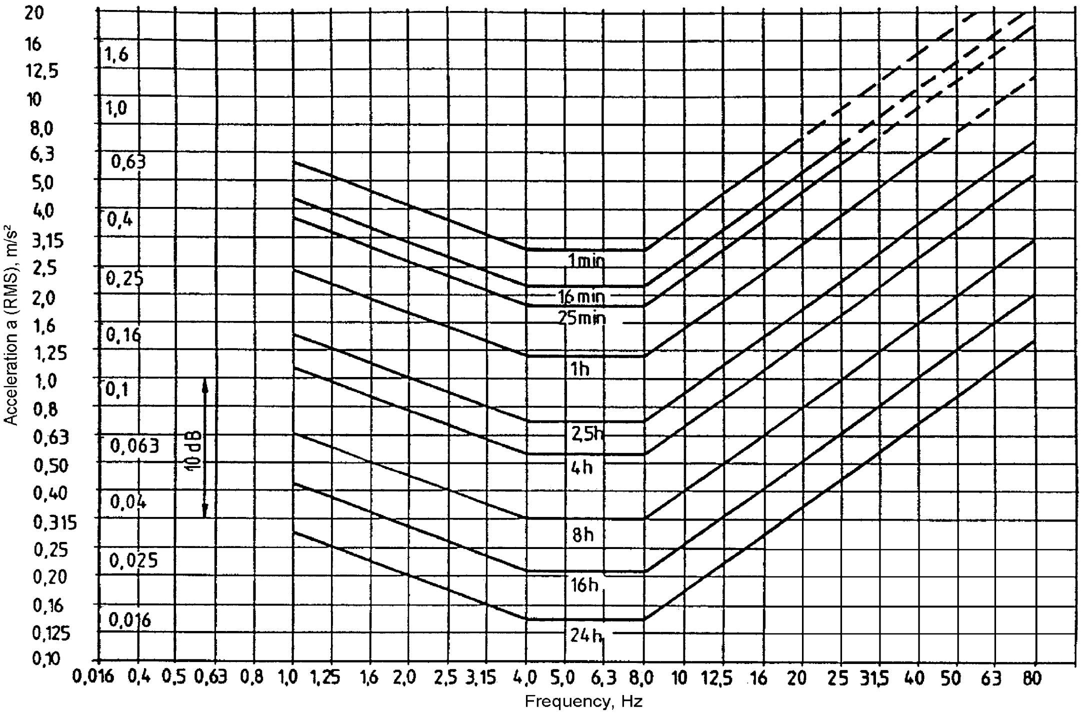

Fig. 1. Fatigue decreased proficiency boundaries for vertical vibrations, ISO 2631.

2.2.2. *VDI norm 2057*

For the vertical vibrations, the Verein Deutscher Ingenieure (VDI) uses the following set of equations as weighting curves for the vibrations and calculates the r.m.s. value of these between 1 and 80 Hz.

$$
\begin{aligned}
1 \leqslant f \leqslant 4\ \text{Hz}, &\quad K_z = 10\,\frac{a_z}{f},\\
4 \leqslant f \leqslant 8\ \text{Hz}, &\quad K_z = 20\,a_z,\\
8 \leqslant f \leqslant 80\ \text{Hz}, &\quad K_z = 160\,\frac{a_z}{f},
\end{aligned}
\tag{2}
$$

where f is the frequency (Hz) and a_z the vertical acceleration (m/s^2).

This value is named the K_z value. Note that the shape of this weighting curve is the same as the filter described in the ISO standard (i.e., the inverse of the curve in Fig. 1, but with a gain of 20). The allowed exposure times for ISO and VDI almost coincide.

The main difference between ISO and VDI is that ISO only refers to the frequency (third band) with the highest weighted value, while VDI takes the total weighted r.m.s. value between 1 and 80 Hz.

2.2.3. *CEN standard*

The proposed standard of the European Committee for Standardisation (CEN) uses a transfer function, given by the following equation, to weight the vibration over the frequency range [3].

The total r.m.s. value of the weighted vibration in the band 1–80 Hz determines the allowed exposure time:

$$|H_z| = \left|\frac{0.42 + 0.045p}{1 + 0.044p + (0.03p)^2}\right| \approx \sqrt{\frac{0.18 + (f/3.54)^2}{[1 - f^2/(8*3.54)]^2 + (f/3.62)^2}}. \tag{3}$$

Notice that the form of this transfer function is almost equal to the inverse of the ISO curve and can be considered a comfort weighting curve where the gain in frequency band between 4 and 8 Hz is taken equal to unity.

The main difference between ISO and CEN is that ISO only refers to the frequency (third band) with the highest weighted value, while CEN uses the total weighted r.m.s. value between 1 and 80 Hz.

2.2.4. *RUG method*

The method used in this investigation was developed at the Laboratory of Vehicle Techniques of the Ghent University (RUG) about 20 years ago. In this method, the comfort is determined by calculating the power spectrum density (PSD) using fast Fourier transformation (FFT) and weighting this by multiplication with the inverse of the ISO weighting curve. The r.m.s. value in the band between 1 and 80 Hz of this weighted PSD curve is used to evaluate the comfort [4,5]. Additionally, the r.m.s. value in each third band is calculated in order to provide a better visual information about the high powered frequencies, which can give an indication about the origin of bad comfort. The method gives results which are similar to the CEN method.

2.2.5. *UCL method*

For the evaluation of shocks the Université Catholique de Louvain (UCL) uses also the r.m.s. value of that part of the ISO-weighted vibration whose amplitude is higher than the so-called 10% level. These are the vibrations higher than a level that is passed more than 10% of the total measuring time.

2.3. *Implementation of the methods*

All the mentioned methods were implemented using Matlab. Starting from a simulated or measured signal the program generates immediately the different allowed proficiency time [5].

3. Measurements

Many measurements were carried out on a forklift (load capacity 2.5 t, diesel) by the UCL (under supervision of Prof. Malchaire from the Institute for Medicine) and are used to compare the results between simulation and reality. The characteristics, i.e., tyre, seat, load, and road profile type, of 16 test runs are summarized in Table 1. On the forklift the vertical accelerations above the front and the rear axle were measured together with the accelerations in three directions

Table 1
Identification of the test runs in Figs. 6, 7 and 11

No	Definition	No	Definition	No	Definition	No	Definition
1	SNPU	5	SAPU	9	RNPU	13	RAPU
2	SNPL	6	SAPL	10	RNPL	14	RAPL
3	SNCU	7	SACU	11	RNCU	15	RACU
4	SNCL	8	SACL	12	RNCL	16	RACL
	Tyre		Seat		Load		Road
S	Solid	N	Normal	L	Loaded	C	Concrete
R	Radial	A	Anti-vibrat.	U	Unloaded	P	Pavement

on the seat. The measurements were recorded on a tape recorder and sampled afterwards in order to calculate the comfort values.

4. Simulation model

4.1. Generalities

Simulations were made using the program SIMULINK, an add-on of the program MATLAB. The total simulation model is split up in subblocks, which can be linked to each other by one line only.

4.2. Wheels and tyres

Because only the vertical vibrations of the wheels were considered in this study, the tyre characteristics used were simplified to a linear parallel damper/spring system. The influence of the nearly rigid fixation of the wheels on the chassis was neglected. Because the vehicle is equipped with relatively small, hard tyres the danger exists that for the rear axle the contact between wheels and ground is lost. For that reason the forces between tyres and ground were always kept positive or zero.

In reality, the values of the wheel damping coefficient and stiffness depend on the frequency, the stroke and the speed of the wheel and can be measured as described in Ref. [6]. Starting from these measurements their characteristics can be implemented as a black box. Nevertheless, in this simulation they were implemented as constant values determined by linearization of the measured values at 6.3 Hz.This frequency value was chosen because measurements on the vehicle showed that the power acting on the wheels was maximal at about 6 Hz.

The lateral stiffness of the tyres was not considered because of the lack of knowledge of the lateral tyre characteristics which are highly non-linear and strongly influenced by the enveloping effect of the tread.

4.3. Chassis

In this simulation the mass of the total forklift was considered as one rigid mass on four spring–damper units, namely the tyres. In this case, all the masses and moments of inertia of the forklift can be reduced to its centre of gravity. Also the vertical input motion for the seat is calculated from this general motion. The vertical displacements of the hubs were calculated from the actual vertical translation and rotations around the horizontal axis through the centre of gravity of the chassis with the general equilibrium equations. For each wheel the force was calculated based on tyre characteristics and on the relative distance and speed between the road surface and the wheel axis. No negative forces were permitted. Horizontal forces were neglected, since this investigation did not deal with the forklift stability, but only with the comfort. The seat was situated at 44% (of the wheel base) behind the front axle and it was assumed that the forks were always the lowest during driving position.

4.4. Seat

Time histories for the input acceleration under the seat and the output acceleration on the seat were available from the measurements. Based on these characteristics, a model for the seat was defined. The seat can be simulated as a black box whose input and output are the mentioned accelerations. This is complicated and difficult work, since the different elements of the seat contain non-linear stiffness and damping characteristics. The "design" of such a black box with the determination of the influence of the frequency, the relative velocity and the displacement is subject of another investigation.

Another possibility is to simulate the seat as a linear system with more degrees of freedom and to optimize a general transfer function between output and input acceleration. For this purpose the general form of the transfer function was written in the following form

$$H(s) = \frac{\sum_{i=1}^{d} a_i s^{i-1}}{\left(1 + \sum_{k=1}^{c} b_k s^k\right)}. \tag{4}$$

In general for the same input, the output calculated by the transfer function has to result in the same value as the output measured during the tests. For that purpose, the frequency domain was divided into 20 classes. For each class the output–input ratio has to be the same for the calculations as well as for the measurements. Because there were 20 classes a transfer function with 20 parameters was chosen and c and d were set to 20 in Eq. (4).

The optimization of these parameters was done with the least-squares method in the Laplace transformed field over the 20 frequency classes. A lot of local minima were found, but no solutions were available which gave a stable transfer function. In all cases, one or more poles were located in the right half-plane.

A third possibility implied a replacement of the seat by a linear model for which the resulting output was sufficiently accurate around its normal working point. This last method was used for the sake of simplicity. As a model for the seat a "two mass–three damper–three spring" system was used as shown in Fig. 2. The different parameter values of the model were calculated via optimization of two seats: a normal one and a commercial "anti-vibration seat".

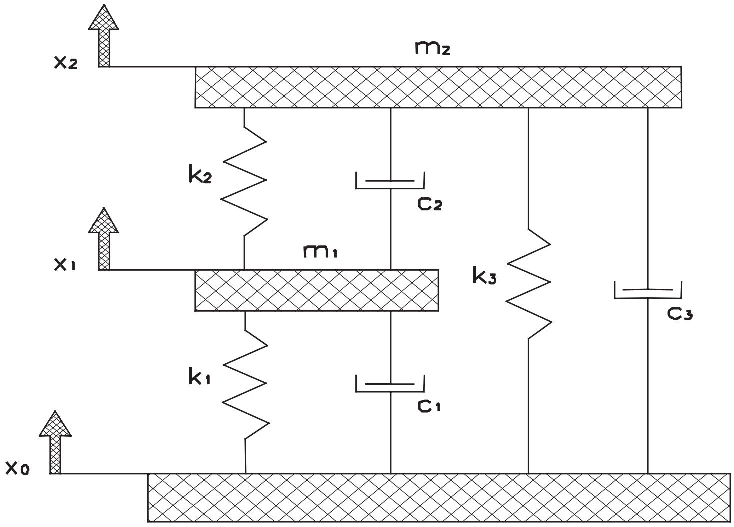

Fig. 2. General scheme of the seat simulation.

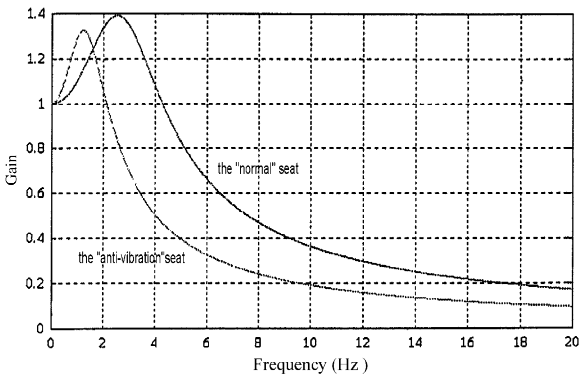

Fig. 3. Bode (amplitude) diagram of the seats.

For this optimization the total frequency band was divided in 20 frequency bands with constant relative bandwidth. In each band the power contents of the ISO-weighted vertical input and output accelerations on the seat were calculated via an FFT and transposed to the centre frequency of the band. For each frequency the frequency response can be written as the amplitude ratio between measurements and simulations. The sum over the 20 different frequencies of the squared differences between the measured and calculated powers was minimized for the eight parameters of the seat model. For the normal seat, three acceptably stable models were found. One of these was selected by comparing the proficiency times resulting from the measurements and the simulation. The form of the used transfer function is given by Eq. (4) with c equal to 2 and d equal to 3. Fig. 3 shows the Bode characteristics for both seats. Fig. 4 shows the comparison between the measured and calculated time histories of the acceleration on the seat for the same input under the seat. For the implementation of the seats a limitation of the negative acceleration to 1 g is included. More details about this linearization can be found in Ref. [5].

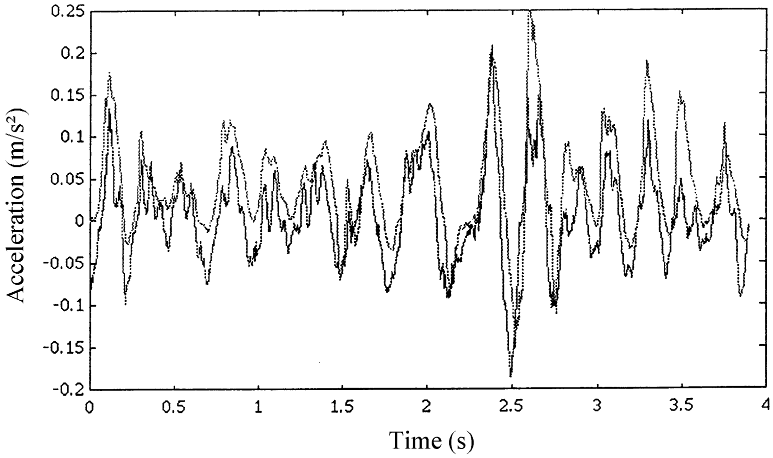

Fig. 4. Measured and simulated acceleration on seat for the same input under the seat.

4.5. Road profile

Three road profiles were simulated: a concrete road, a block road and a block profile typical for a forklift.

The road profiles were described by their power spectrum density $D(\Omega)$ as shown in Fig. 5 and were linearized using the following equation, in which k was set at 2 and the road pulsation $\Omega_0 = 1/(2\pi)\,(1/\mathrm{m})$ as foreseen in the ISO 8608 standard [7].

$$D(\Omega) = D(\Omega_0)\left|\frac{\Omega}{\Omega_0}\right|^{-k}. \tag{5}$$

One of the practical advantages of taking k equal to 2 is that the first derivative of the road profile amplitude will be an integrated white noise or a pink noise (constant power in constant relative bandwidth). The first derivative equals the slope of the profile in the distance domain or the velocity of the profile in the time domain.

To define the other parameter in this equation, effective measurements on the test roads were performed and the value of $D(\Omega_0)$ was optimized in such a way that the r.m.s. values for the vertical acceleration of the centre of gravity were the same on the simulated as well as on the effective road. This method can be used since the total model was linearized and the output parameters depend only on the input amplitude of the pink noise and his realization.

For the simulation the profile amplitude input for the left hand front wheel was a time integrated white noise. The right hand front wheel is rolling over exactly the same profile but with a time delay corresponding to the speed and the distance between the right hand front and left hand rear wheels. With this expression the coherency between the two tracks is respected [4]. For the rear wheels the same road profiles were used as for the respective front wheels but with a distance delay corresponding to the wheel base.

For the typical block profile a plank of 1-in height and 8-in width was adopted; this is a very common test for forklifts.

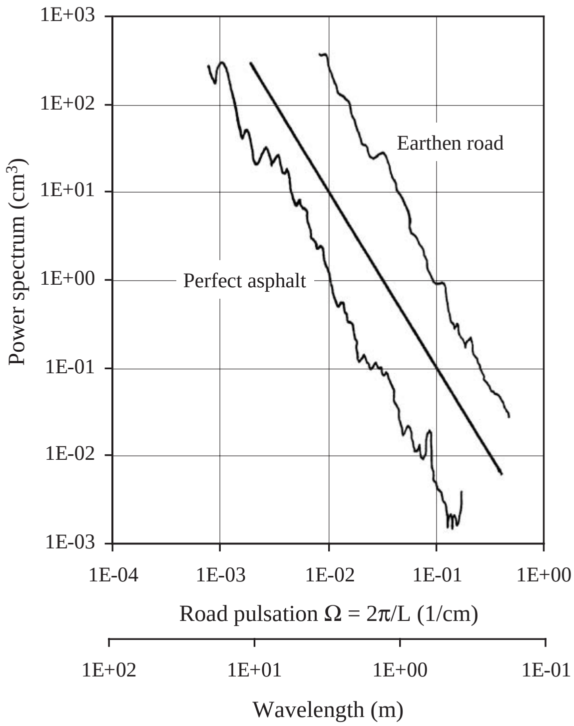

Fig. 5. Road profile characterization.

5. Results

5.1. Simulation of the test runs

In order to validate the simulation model, the 16 test runs as indicated in Table 1 were simulated and their vibrations in the vertical direction were analyzed. For the same combinations of tyres, seats, road profiles and loads the results of the simulations and the measurements were compared. The comfort results are given in Fig. 6. The correlation between measurement and experiment is satisfactory. Not only the relative ranking is nearly the same, but also the absolute value.

The largest deviations occurred for the anti-vibration seat, since the seat was simulated as a linear system, while in reality the seat behaves non-linearly.

5.2. Influence of shocks

The influence of the shocks was measured according to the UCL method. For the 16 test runs as defined in Table 1, Fig. 7 shows the r.m.s. value of the total run and the r.m.s. value of the so-called "10% vibrations". The values of the last ones were approximately twice these of the first ones. The influences of the different parameters described hereafter on the normal comfort values are analogous with the influences on the 10% values.

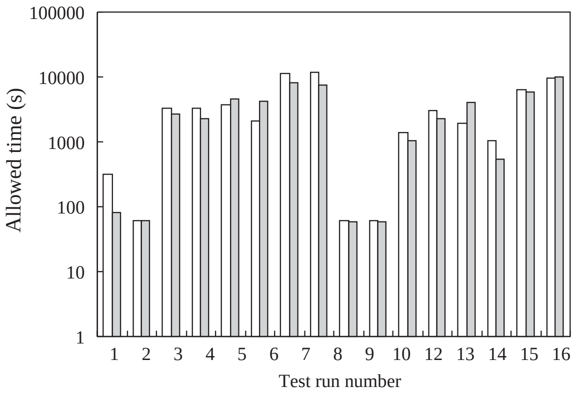

Fig. 6. Comparison between (white bar) simulations and (grey bar) test runs (VDI method).

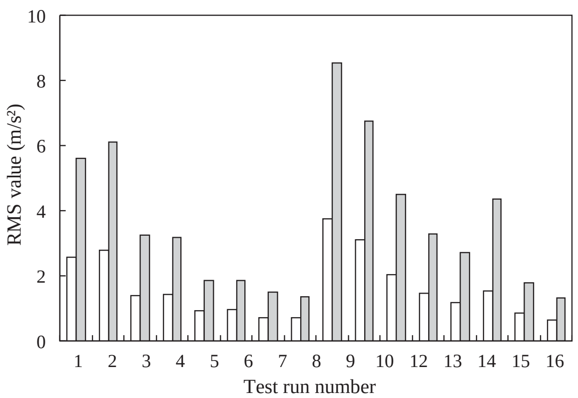

Fig. 7. Comparison between test runs (UCL method); (white bar) r.m.s.; (grey bar) 10% r.m.s.

5.3. Measurement in three directions

In many reports about comfort in vehicles it is convenient that only the vertical vibration on the seat is measured and evaluated. For vehicles with a large wheel base this is mostly acceptable, but for short ones as a forklift, it is not evident. Fig. 8 shows the measurement results for the anti-vibration seat when only the vertical vibrations are evaluated.

When the horizontal vibrations are also taken into account, one obtains Fig. 9. The difference in allowed time is relatively important. The explanation for this can be the influence of the short wheel base, but also the enveloping effect of the relatively small tyres.

From the simulation the horizontal component from the rolling and pitching motions excited by the small wheel base and track width can also be calculated. When comparing the measurement and the simulation results, Fig. 10, it can be concluded that approximately 60% of the difference comes from these motions. The rest is possibly due to the enveloping effect of the tyre, but this is part of current investigation.

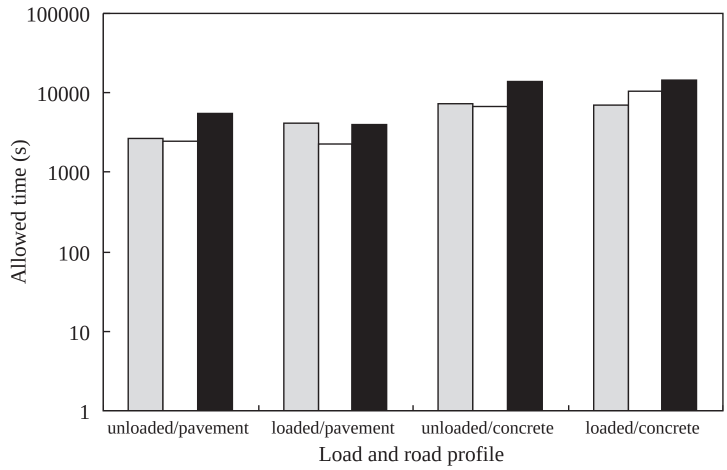

Fig. 8. Allowed proficiency time derived from *X* direction—measurements; (grey bar) diagonal tyre; (white bar) radial tyre; (black bar) solid tyre.

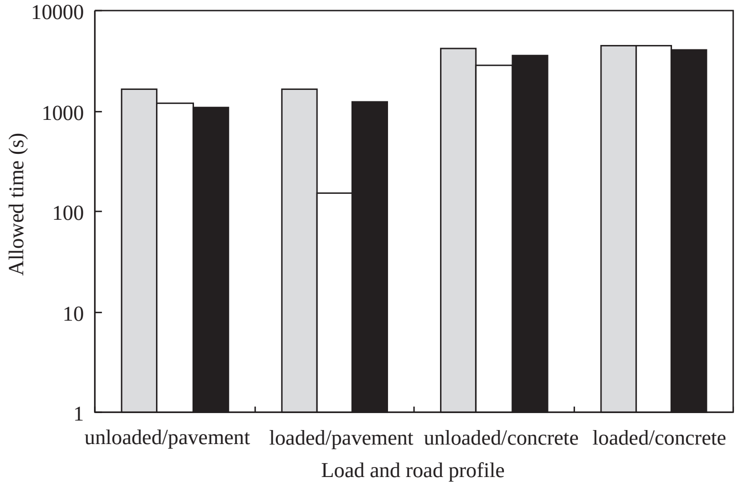

Fig. 9. Allowed proficiency time derived from three directions—measurements; (grey bar) diagonal tyre; (white bar) radial tyre; (black bar) solid tyre.

5.4. Correlation between the different evaluation methods

Roughly, one can say that the ranking of the allowed time for the different test runs was the same for all evaluation methods. Of course, the absolute values are different. Calculated over 24 measurements, the mean value for the ratio of allowed ISO time and VDI time is 1.6 with a minimum of 1.1 and a maximum of 2.1. However, this is not a big problem because for each method the authors declare that the allowed time can be influenced by different parameters such

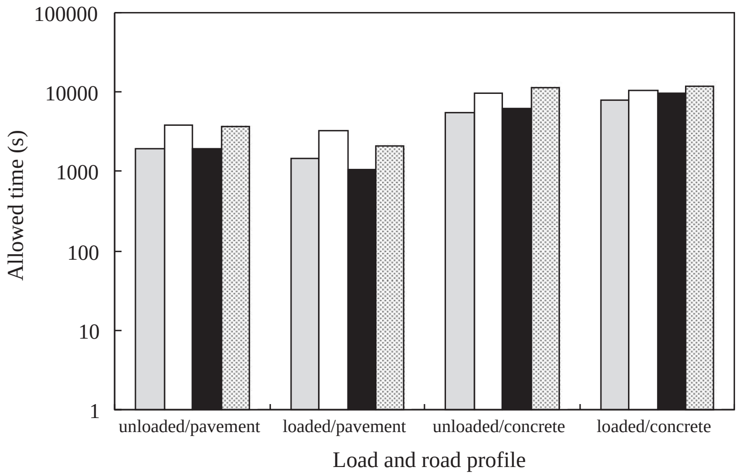

Fig. 10. Comparison between allowed proficiency evaluated from 1 and 3 directions—simulations; (grey bar) radial tyre XYZ; (white bar) solid tyre XYZ; (black bar) radial tyre X; (spotted bar) solid tyre X.

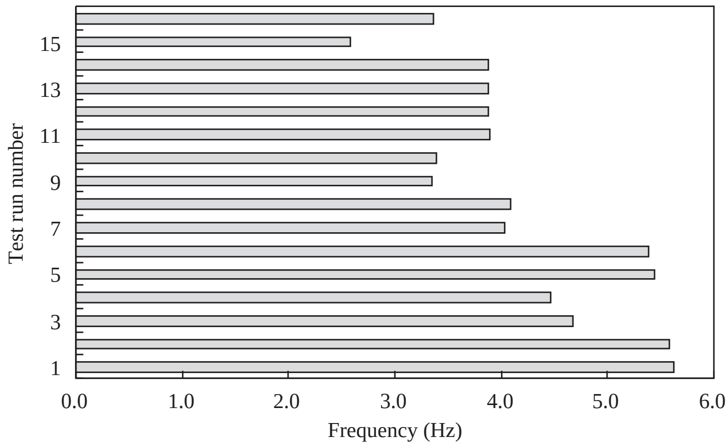

Fig. 11. Frequency of the peak value in the spectrum—measurements.

as the work environment, the age, etc. In the case of a strict legislation the method has to be exactly defined. The differences between the VDI, the CEN and the RUG method were very small.

5.5. Peaks in the power spectrum

The frequencies where the peaks in the power spectrum occurred are shown in Fig. 11 for the runs defined in Table 1. These plots are based on measurements. An analogous figure based on the simulations gave the same results (data not shown). The length of the horizontal bars is a measure for this frequency. For solid tyres (runs 1 to 8) the frequencies were higher than for

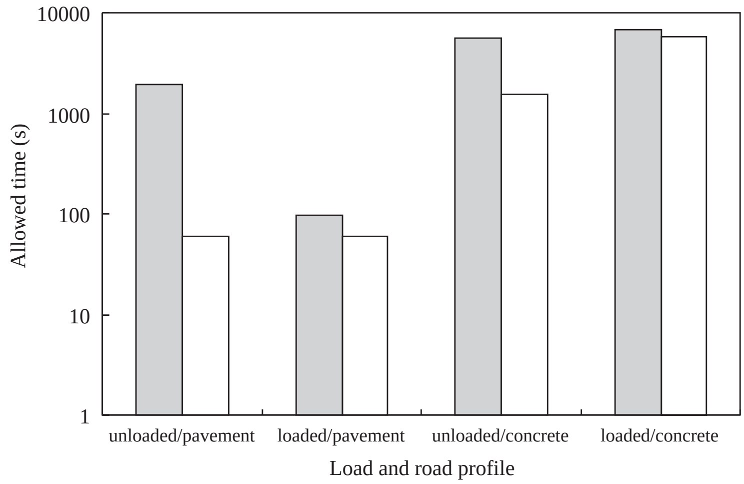

Fig. 12. Influence of the tyre type on the allowed fatigue time with a normal seat—simulations; (grey bar) solid tyre; (white bar) radial tyre.

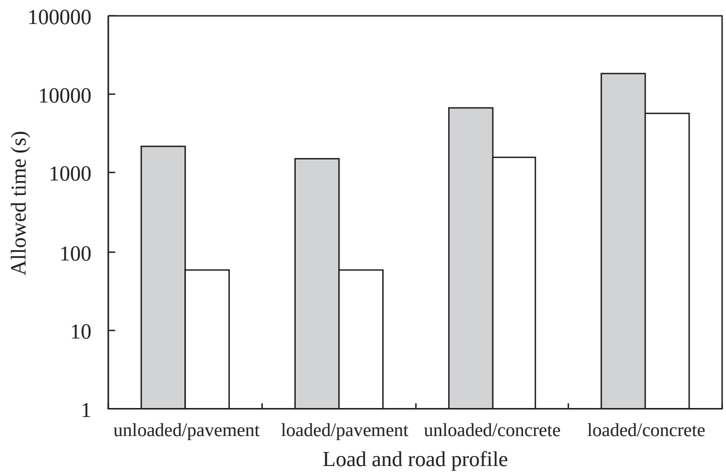

Fig. 13. Influence of the seat type on the allowed fatigue time—simulations; (grey bar) anti-vibration seat; (white bar) normal seat.

pneumatic tyres. The equal-comfort curves have a minimum between 4 and 8 Hz (Fig. 1). So, normally the comfort can be expected to be smaller for solid tyres. This also means that the influence of the load on the comfort will be higher for pneumatic tyres than for solid ones.

5.6. *Influences*

Fig. 12 shows the influence of the tyres on the fatigue limit for the simulations with the normal seat. The influence of the tyres is clear. With solid rubber tyres the time allowed was longer than with pneumatic tyres. Especially for an unloaded forklift on concrete the difference was very high.

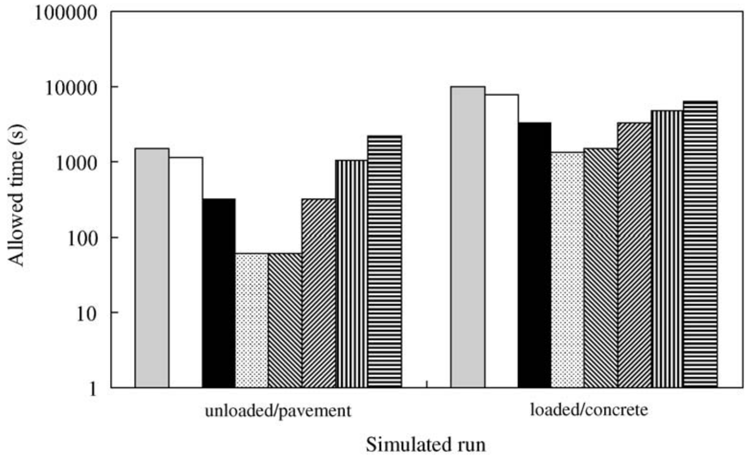

Fig. 14. Influence of the tyre characteristics on the allowed proficiency time—simulations; (bars from the left to the right) $k = 0.5$; 0.7; 1.0; 2.0; $c = 0.5$; 1.0; 1.5; 2.0.

With the anti-vibration seat the results were analogous, but the differences decreased. This can be seen in Fig. 13: the influence of the seat was more important than that of the tyres.

From the previous one should expect that the comfort with pneumatic tyres was higher than with solid ones, but this did not result from the measurements or the simulations, nor from the declarations of the drivers. The reason for this discrepancy can be explained by the unweighted r.m.s. values of the vertical vibrations. The values for the solid tyres were smaller than for the pneumatic ones. Reason for this is the damping, which is six times higher for solid tyres. Fig. 14, on which the influence of tyre stiffness and damping is very clear, results from simulations with a normal seat.

A disadvantage of solid tyres is the higher damping which gives rise to higher rolling resistance, which implies that the power of the engine has to increase. The advantage of the higher stiffness, however, is the stability of the vehicle. When heavy loads are lifted up, the lateral and longitudinal displacement of the centre of gravity will be higher with soft tyres, which can lead to instability.

6. Conclusions

The different evaluation methods used to determine the comfort give very analogous results, the absolute values differed slightly but the ranking was always the same.

The analysis of simulated and measured vibrations showed that solid tyres gave better comfort than pneumatic ones because the positive influence of the higher damping was greater than the negative influence of the higher stiffness. The seats, however, were found to have the most important influence. The correlation between the measurements and the simulations was satisfactory, so the model can be used for further investigations on the influence of other parameters on the vehicle.

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