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Approaches for a 3D assessment of pavement evenness data based on 3D vehicle models

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A R T I C L E   I N F O

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A B S T R A C T

Pavements are 3D in their shape. They can be captured in three dimensions by modern road mapping equipment which allows for the assessment of pavement evenness in a more holistic way as opposed to current practice which divides into longitudinal and transversal evenness. It makes sense to use 3D vehicle models to simulate the effects of 3D surface data on certain functional criteria like pavement loading, cargo loading and driving comfort. In order to evaluate the three criteria mentioned two vehicle models have been created: a passenger car used to assess driving comfort and a truck-semitrailer submodel used to assess pavement and cargo loading. The vehicle models and their application to 3D surface data are presented. The results are well in line with existing single-track (planar) models. Their advantage over existing 1D/2D models is demonstrated by the example of driving comfort evaluation. Existing “geometric” limit values for the assessment of longitudinal evenness in terms of the power spectral density could be used to establish corresponding limit values for the dynamic response, i.e. driving comfort, pavement loading and cargo loading. The limit values are well in line with existing limit values based on planar vehicle models. They can be used as guidelines for the proposal of future limit values. The investigations show that the use of 3D vehicle models is an appropriate and meaningful way of assessing 3D evenness data gathered by modern road mapping systems.

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1. Introduction

Pavements are 3D in their shape. However, current practice in most countries is to measure and assess pavement evenness in terms of two dimensions: the longitudinal and transversal profile. The only reason for this is a technical one: the measurement equipment has not been sufficient enough in the past to gather 3D road surface data with the efficiency and accuracy needed to assess pavement evenness.

In recent years measurement techniques have come up, for the first time, to allow for an effective and sufficiently accurate acquisition of 3D road surface data. On one hand this represents a challenge; on the other hand it opens opportunities of assessing pavement evenness in a more comprehensive and realistic way in the future. This paper covers...
approaches for a 3D assessment of pavement evenness data. Basically, there are two opposing approaches to assess pavement evenness, one evaluating the dynamic effects of the road on passengers, vehicles and the road itself; the other one focusing on the evaluation of the geometrical properties of the pavement surface. The first approach has been chosen for the 3D assessment of pavement evenness and presented in the paper. To begin with, a short review on existing approaches to evaluate longitudinal evenness shall be given.

2. Existing approaches to evaluate longitudinal evenness

Existing approaches to the evaluation of longitudinal evenness may roughly be divided into equipment-specific methods and numerical methods based on the measured longitudinal elevation profile (Fig. 1). Equipment-specific methods include, for example, rolling straightedge, slopemeter and profilograph.

For monitoring purposes on a net-wide level, measuring the “true” longitudinal elevation profile using a non-contacting profilometer is preferred, followed by the calculation of suitable indicators of longitudinal evenness from the measured profile. These can be either “geometrical” indices calculated directly from the elevation profile or its 1st, 2nd (Bruscella et al., 1999; Hudson et al., 1985; Rouillard et al., 2000) and even 3rd (Schniering, 1998) derivatives respectively, e.g. mean, median, standard deviation, root mean square, variance, range, etc., or indices inferred indirectly by means of wavelet decomposition (Shirakawa et al., 2005, 2006; Wei et al., 2005), and Fourier transforms (Andrén, 2006; Braun, 1969; Houbolt, 1962; Sayers et al., 1986). Besides that diverse filtering techniques (moving average, Butterworth, Chebyshev) are used to pre-process the profile data and to calculate unevenness indices with respect to different wave bands (e.g. short, medium and long waves) as described in prEN 13036-5 (2004).

An alternative is to deduce the dynamic response of measuring devices or vehicle components (axles, bodywork, seats, and cargo load) and/or the perception of driver/passengers from the measured elevation profile by appropriate filters and to express the output in terms of indicators giving a statistical and/or peak rating for a given evaluation length (response-type indicators). Approaches of this kind include, amongst others, the international roughness index (IRI) (Sayers, 1986, 1996), the half-car roughness index (HRI) (Sayers, 1989), even a full-car roughness index (Capuruco et al., 2005), and the ride number (RN) (ASTM E 1489-98, 2003), which is defined as an exponential transform of the profile index (PI). The profile index, in turn, uses the same quarter car filter as the IRI, but with other coefficients.

Fig. 1 – Existing approaches for the evaluation of longitudinal evenness. (a) Geometry in distance domain (longitudinal profile). (b) Geometry in spectral domain. (c) Effect in distance domain. (d) Geometry and effect in distance domain.
ride number is a “comfort” indication number, scaled from 5 (perfectly smooth) to 0 (the maximum possible roughness) and based on human rating experiments.

A more recent approach of human rating experiments uses fuzzy set theory (Loizos, 2001). The results were compared to IRI measurements. Another example of a response-type indicator is the dimensionless “effective evenness index” (LWI) (Ueckermann, 2002). The LWI uses 3 different vehicle filters along with a “human perception” filter (ISO 2631-1, 1997) and incorporates the effects of longitudinal unevenness on comfort, driving safety, pavement loading, and loading of freight cargoes, the greatest of them within an evaluation section being determent for the LWI.

One of the drawbacks of response-type indicators as compared to geometrical indicators is that they are linked to specific speeds and system properties, so that evaluation may not be sufficiently comprehensive or objective. It can, however, be argued that the point of interest for the evaluation is not the geometrical shape of the road surface, but its dynamic effects. The advantage of the response-type indicators is that they permit differentiated evaluation of longitudinal unevenness with respect to irregular, periodic, and local characteristics.

Some of the above mentioned evaluation approaches are used in a more academic environment, others are widely used in pavement monitoring practice on a national or even international level. Diverse national and international experiments have been conducted in the past in order to correlate and harmonize evenness measuring and evaluation methods, like the international road roughness experiment (IRRE) (Sayers, 1986), the European FILTER experiment (Alonso and Yanguas, 2001), and the PIARC EVEN experiment. A recent comparison between profile index (PI) and international roughness index (IRI) can be found (Wilde et al., 2007). For a comparison of three widely used evenness evaluation methods in Europe, IRI, power spectral density (PSD) and wave bands analysis (Delanne and Pereira, 2001), a rather comprehensive table of roughness devices and indicators has been put together by Praticò (2004) and Boscaino et al. (2004).

The method presented in this paper is a response-type approach evaluating the dynamic effects of the road on passengers, vehicles, and the road itself.

3. Spectral description of longitudinal unevenness

Fig. 2 shows the spectra of five different road surfaces. The ordinate plots the PSD of the longitudinal elevation profile—a spectral quantity which is proportional to the square of the amplitudes of the wavelengths represented in the elevation profile. The abscissas indicate the spatial angular frequency and the corresponding wavelength. The wavelengths shown in the graph cover a band of approximately 0.1—100 m.

It is evident from the chart that long waves are associated with high amplitudes and short waves are with low amplitudes. Plotted in double logarithmic scale, these power density spectra can approximately be described by a straight line with a gradient between −1.5 and −3. The negative gradient is referred to as “waviness”($w$), while the value of the straight line at a wavelength of 6.28 m (2π m) denotes the “unevenness index” of the road (ISO 8608, 1995; prEN 13036-5, 2004). In the present context it will be denoted as $G_0$. A $G_0 = 1$ cm$^2$ marks a good motorway and a $G_0 = 9$ cm$^2$ marks a bad motorway.

Longitudinal unevenness may be sub-divided into irregular unevenness, periodic unevenness (periodicities), and single (transient) occurrences. The benefit of the PSD approach is that it provides a description of the surface characteristics in terms of two dimensions, wavelength and amplitude. However, since it lacks the phase information it is only able to describe the evenness in a statistical manner. The information about location and shape of the unevenness gets lost. Thus, the PSD is best suited for the description of the irregular, overall characteristic of the unevenness. We will use the PSD later on for the deduction of limiting levels for the proposed evenness indicators.

4. The effective evenness index

The effective evenness index LWI consists of a set of filters which is applied to a longitudinal profile in order to calculate the dynamic response in terms of axle loading, cargo loading, and perceived vertical loading on a passenger car seat (Ueckermann, 2002). It is based on planar, 2D vibration models using only one longitudinal profile. As such it is a sheer 2D model, covered here only for comparison with the results calculated from the 3D models presented.

Fig. 3 illustrates the 3 assessment criteria represented by the LWI: 1) the axle load of a 11.5 t axle calculated from a two-mass system “driven” with 80 km/h over the profile; 2) the vertical acceleration on the loading area of a 3 axle semitrailer calculated from a plane four-mass system “driven” with 80 km/h along the profile; 3) the perceived vertical acceleration on the seat of a passenger car “driven” with 100 km/h over the profile, represented by a three-mass quarter-car system. The LWI calculation scheme is shown in Fig. 4.

There is a theoretical relationship between the three filter responses of LWI and the power spectral density in terms of its parameters $G_0$ and $w$ which are mentioned next for sake of completeness. Mitschke and Wallentowitz (2000) showed that the variance $\sigma_q^2$ of a vehicle’s response $q$ (i.e. axle loads or
accelerations) related to the power spectral density according to

\[ s^2 = G_0 \omega^{\nu-1} \Omega_0 \int \frac{V^2(\omega)}{\omega^\nu} d\omega \]

where \( v \) denotes the velocity of the vehicle, \( \Omega_0 \) is a constant (reference spatial angular frequency), \( \omega \) is the angular frequency, \( V^2(\omega) \) is the transfer function of that particular vehicle response.

5. Vehicle models for assessment of 3D evenness

For the assessment of 3D evenness two vehicle models have been set up using MATLAB code: one 3D passenger car model pictured in Fig. 5 and one 3D truck-semi trailer model. The passenger car model features 6 masses representing the four wheels, the car body, and the driver. The masses are connected to each other by spring-damper units. Additionally, two anti-roll bars are placed between the wheels of the left and right side of the car in order to limit the rolling movement of the car. The model is intended to simulate the vertical dynamics of a car, i.e. vertical movements, rolling, and pitching. Fig. 6 shows the transfer functions of the dynamic wheel loads for the two front wheels. The amplitude response is shown in Fig. 6(a) and (c), the phase response in Fig. 6(b) and (d). The solid lines mark the responses by excitation over the left wheel track and the dash lines mark the responses by excitation over the right wheel track. Two peaks can be found in the amplitude responses: one at about 1–2 Hz and one at about 10–15 Hz. They mark the eigenfrequencies of the car body and car axle respectively.

Fig. 7 shows the transfer functions of the driver’s vertical acceleration. Fig. 7(a) and (b) were values measured on the seating surface and Fig. 7(c) and (d) were values perceived by the driver. The amplitude responses are shown in Fig. 7(a) and (c), the phase responses are in Fig. 7(b) and (d). The amplitude response of perceived acceleration is calculated from the amplitude response of acceleration by multiplying it with the frequency weighting curve given in ISO 2631-1 (1997) (Fig. 8). This is to account for the frequency-dependent perception of the driver. Again, the solid lines mark the responses by excitation over the left wheel track and the dash lines mark the responses by excitation over the right wheel track. Several local maxima can be seen from the amplitude responses (e.g. at about 1, 10 and 20 Hz) as well as local minima in those frequencies. They result from the fact that there are certain frequencies arising from the wheel base of the car and the driven velocity, which lead to a in-phase or opposite-phase excitation of the front and
rear axles, and in turn lead to the corresponding maximum or minimum acceleration in vertical displacement of the driver on the seat.

In Fig. 9 the truck-semi trailer model is shown. It is a submodel consisting of the rear part of the truck frame featuring an 11.5 t axle equipped with dual tyres, and the semi trailer equipped with 3 axles each having two wheels and covering a load of 7.3 t. The axles are attached to the vehicle frames by springs, dampers, and anti-roll bars. The geometric dimensions as well as the springs and dampers

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**Fig. 6** – Transfer functions of dynamic wheel loads. (a) Amplitude responses for left front wheel. (b) Phase responses for left front wheel. (c) Amplitude responses for right front wheel. (d) Phase responses for right front wheel.

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**Fig. 7** – Transfer functions of driver vertical acceleration. (a) Amplitude responses of vertical acceleration. (b) Phase responses of vertical acceleration. (c) Amplitude responses of perceived acceleration. (d) Phase responses of perceived acceleration.
constants are taken from technical drawings and data sheets obtained from truck and trailer manufacturers (Mercedes Actros, Koegel Eurotrailer, BPW Bergische Achsen). The center of gravities and moments of inertia are taken from Bachmann et al. (2008).

The body of the semitrailer is attached to the truck frame by a ball joint representing the fifth wheel coupling. For the purpose of simplification this joint is centered above the dual-wheel axle. Another simplification of the model is that the body of the semitrailer is stiff regarding bending and torsion. The model represents a standard European semitrailer with a wheel base of 1.31 m, a distance between center axle and fifth wheel coupling of 6.39 m and an overall length of 13.62 m.

Fig. 10 shows the transfer functions of the dynamic wheel loads of the 11.5 t truck axle for pure vertical excitation. The designations Ra, Ri, Li and La denote the four wheels of the truck axle and are identified in Fig. 9. The amplitude responses are shown in Fig. 10(a) and (c), the phase responses are in Fig. 10(b) and (d). Two eigenfrequencies can be identified from the plots: one at about 6 Hz for the rear body of the truck and one at about 17 Hz for the axle. Compared with Fig. 10 the eigenfrequencies are shifted towards higher frequencies because of the additional influence of the anti-roll bar. It can be seen from the plots that there is a small “dynamic” force remaining at “zero” frequency. This is due to the twisting of inner and outer wheels against each other when being moved up or down by the same amount during rolling.

Fig. 11 shows the transfer functions of the dynamic wheel loads of the 11.5 t truck axle for rolling excitation (opposite-phase excitation in the right and left wheel tracks by moving the right-hand side wheels and left-hand side wheels by the same amount but in opposite directions). The amplitude responses are shown in Fig. 11(a) and (c), the phase responses are in Fig. 11(b) and (d). Two eigenfrequencies can be identified from the plots: one at about 1 Hz for the rear body of the truck and one at about 11 Hz for the axle. Compared with Fig. 10 the eigenfrequencies are shifted towards higher frequencies because of the additional influence of the anti-roll bar. It can be seen from the plots that there is a small “dynamic” force remaining at “zero” frequency. This is due to the twisting of inner and outer wheels against each other when being moved up or down by the same amount during rolling.

Fig. 12 depicts the transfer functions of the vertical acceleration on the cargo area of the semitrailer above the center axle. The amplitude responses are shown in Fig. 12(a) and (c), the phase responses are in Fig. 12(b) and (d). Fig. 12(a) and (b) apply to the semitrailer: the solid line marks the excitation by the three wheels of the right-hand side and the dash line marks the excitation by the three wheels of the left-hand side of the semitrailer. They are the same since the semitrailer and its cargo load are assumed to be symmetric about its longitudinal axis. Fig. 12(c) and (d) apply to the truck axle: four curves are displayed in the graphs representing the four wheels (Ra, Ri, Li, and La). Since the ball joint is relatively far away from the point on the cargo floor where the acceleration is determined the influence of the truck axle on that particular point is relatively small compared to the influence of the trailer axles which can be seen from the different scales of the plots (compare Fig. 12(c) and (d) with Fig. 12(a) and (b)).

Eigenfrequencies of the truck at about 1 Hz for the body and about 10 Hz for the axle can be seen from Fig. 12(c). The amplitude response of the semitrailer (Fig. 12(d)) reveals local maxima and minima. The reason for this is similar to the reason given for the amplitude response of the driver acceleration (Fig. 7). A first maximum response can be found at about 2 Hz.

6. First plausibility checks of the 3D models

In the following figures the result of first plausibility checks of the MATLAB model are illustrated. Fig. 13(a) presents the wheel loads of the 11.5 t axle when driven with 14 km/h over an 8 cm high and 3 m long bump with a sinusoidal shape. The solid line represents the calculated wheel load and the dash line represents the measured wheel load. As can be seen from the different scales of the solid line and the dash line the static axle loads were different. However, the relevant information about the vibrational behavior lies in the shape rather than in level of the curve. Regarding the shape of a close match between calculated and measured responses can be attested, which confirms that the dynamic properties of the model are comparable to real axles.
In Fig. 13(b) a comparison of wheel loads calculated by MATLAB model and a rather complex truck model created with the multi-body software MSC.ADAMS/Car is shown. The solid line marks the simpler MATLAB model whereas the dash line displays the result of the ADAMS model. Despite the fact that the static wheel loads differ due to different cargo loads assumed, the comparison reveals a close agreement between the dynamic properties of the

Fig. 10 – Transfer functions of dynamic wheel loads for vertical excitation. (a) Amplitude responses for right wheels. (b) Phase responses for right wheels. (c) Amplitude responses for left wheels. (d) Phase responses for left wheels.

Fig. 11 – Transfer functions of dynamic wheel loads for rolling excitation. (a) Amplitude responses for right wheels. (b) Phase responses for right wheels. (c) Amplitude responses for left wheels. (d) Phase responses for left wheels.
models. Both the measured data and the results of the ADAMS model shown in Fig. 13 are taken from Bachmann et al. (2008).

Fig. 14(a) shows the dynamic wheel loads for the 11.5 t truck axle on a bad motorway as calculated by the MATLAB model. Ra, Ri, Li and La denote the 4 wheels of the axle. The velocity is assumed to be 85 km/h. The 3D surface is generated by a MATLAB program which allows to create realistic road surfaces based on a specified power spectral density and spatial coherence function which describes the coherence of parallel wheel tracks as a function of the spatial angular frequency (or wavelength) and their distance to each other. Fig. 14(b) shows the corresponding dynamic wheel loads for the semitrailer center axle. 2R and 2L denote the wheel positions. The dynamic wheel load is expressed in percent of static wheel load. The dynamic load coefficient (DLC) describing the quotient of root-mean-square and static value is 17.8% for the truck axle and 19.0% for the semitrailer axle.

Table 1 summarizes the results of the dynamic wheel loads in terms of the DLC for the 11.5 t axle with respect to three different levels of evenness and three different levels of model complexity represented by the multi-body MSC.ADAMS model, the 3D MATLAB model presented here, the 1D pavement loading filter of the effective evenness index (LWI), and a two-mass model addressed in Section 4. The calculations are based on a velocity of 85 km/h. The data from MSC.ADAMS are taken from Bachmann et al. (2008). As seen from Table 1, close agreement can be found between the results of the three models confirming the suitability of the 3D MATLAB model presented here.

In order to understand the numbers presented in Table 1 and to widen the scope to cover the three assessment...
criteria addressed in Section 4, Table 2 is presented which illustrates the impact of a bad evenness on pavement loading, cargo loading and human exposure to vibration. According to Table 2, which is based on the three filters of the effective evenness index (LWI), a motorway in bad condition \((G_0 = 9 \text{ cm}^3)\) creates an increase in pavement loading of up to 50% and in cargo load acceleration of up to \(3 \text{ m/s}^2\). This is a rather high value as 90% of accelerations, which found on cargo floors of semitrailers are below this figure according to a German standard (DIN 30786-2, 1986).

Furthermore, the driver would be exposed to a perceived acceleration of \(0.9 \text{ m/s}^2\) (rms) which, according to ISO 2631-1 standard (1997), would be “extremely noticeable” and would create a “substantial health risk” if the driver would be exposed to it 8 h per day over a longer period of time.

To conclude the section of plausibility checks a run of the 3D truck-semitrailer with 85 km/h over a generated road of medium evenness featuring a sinusoidal bump (1 m long and 3 cm tall) has been simulated. The results can be seen in Fig. 15, \(G_0 = 2.2 \text{ cm}^3, w = 2, v = 85 \text{ km/h}\). Fig. 15(a) illustrates the dynamic wheel loads of the truck axle with its 4 wheels. Fig. 15(b) illustrates the dynamic wheel loads of the 3 semitrailer axles. The blue curves mark the first, the green ones the second, and the red ones the third axle of the trailer. In Fig. 15(c) the vertical acceleration (cargo load acceleration) of the semitrailer above its center axle is shown. It can be seen that the dynamic wheel load of the truck axle reaches the static wheel load at the position where the bump is located, i.e. the axle is about to lift off. The dynamic wheel loads of the semitrailer axles shown in Fig. 15(b) are smaller and reach only about 80% of their static loads at that very point. The reason is that they are equipped with a comparably “softer” suspension system resulting from typical design constraints. Note the three peaks slightly displaced against each other corresponding to the wheel base of the axles (1.31 m). The same observation can be made in Fig. 15(c) representing the cargo load acceleration: three short strokes can be observed at the location of the bump corresponding to the three axles passing that point. The oscillation with a period length of about 10 m following the three pulses corresponds to the first maximum of the amplitude response function at about 2 Hz shown in Fig. 12.

7. Application to real 3D road surfaces

For the application of the 3D models we could resort to 3D surface data measured by a mobile mapping system featuring a rotating laser in combination with an inertial measuring unit and a global positioning system. In a first step the assumed wheel tracks were extracted from the geometric surface data—two wheel tracks for the passenger car, two wheel tracks for the semitrailer and four wheel tracks for the truck axle. The passenger car was assumed to pass the road at 100 km/h. For the truck-semitrailer a speed of 85 km/h was applied. Three evenness criteria were evaluated: the vertical

### Table 1 – Dynamic wheel load coefficients.

<table>
<thead>
<tr>
<th>Evenness ((w = 2))</th>
<th>MSC.ADAMS</th>
<th>MATLAB 1D/2D model (LWI)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Good ((G_0 = 1 \text{ cm}^3))</td>
<td>5.3%</td>
<td>5.9%</td>
</tr>
<tr>
<td>Medium ((G_0 = 3 \text{ cm}^3))</td>
<td>9.3%</td>
<td>10.1%</td>
</tr>
<tr>
<td>Bad ((G_0 = 9 \text{ cm}^3))</td>
<td>17.4%</td>
<td>17.8%</td>
</tr>
</tbody>
</table>

### Table 2 – Impacts of evenness on pavement, cargo and driver.

<table>
<thead>
<tr>
<th>Evenness ((G_0))</th>
<th>Forces exerted on pavement: driving safety</th>
<th>Payload acceleration</th>
<th>Forces exerted on human body</th>
</tr>
</thead>
<tbody>
<tr>
<td>(G_0 \text{ (cm}^3))</td>
<td>Max/rms increase/decrease in wheel load (with respect to static wheel load) (%)</td>
<td>Max/rms vertical acceleration on the loading area (m/s²)</td>
<td>Effective value of the frequency-weighted vertical acceleration (m/s²)</td>
</tr>
<tr>
<td>1</td>
<td>18/6</td>
<td>1.0/0.3</td>
<td>0.3</td>
</tr>
<tr>
<td>3</td>
<td>30/10</td>
<td>1.7/0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>9</td>
<td>50.0/16.5</td>
<td>3.0/0.9</td>
<td>0.9</td>
</tr>
</tbody>
</table>
perceived acceleration of the driver in the passenger car, the vertical acceleration on the cargo area of the semitrailer above the center axle and the dynamic wheel loads of the truck axle.

Fig. 16 presents the results for a road section offering a good evenness. The section length is 100 m. Four graphs are shown in the figure: the wheel tracks which have been extracted from the surface data can be seen in Fig. 16(a), followed by the cargo load acceleration, the dynamic wheel loads and finally the acceleration the driver perceives. The results are given in terms of rms acceleration or DLC which is the rms of the dynamic wheel load divided by the static wheel load. In the corresponding graphs they are displayed by the blue curves. For comparison purposes the results of the respective 1D/2D models are given as well displayed by red curves. They are based on the three filters of the effective evenness index (LWI) which address the same evenness criteria and same speeds that are used for the 3D models. The 1D/2D planar models use the right wheel track only which is denoted by “R” in the case of the passenger car and the semitrailer and by “Ra” in the case of the truck axle. The results of both 3D model and planar model, comply well with the values given in Table 2 for a “good” road. Good agreement between the 3D model and the planar 1D/2D model can be observed with one exception: the rms acceleration of the driver in the planar model is about twice the magnitude of the 3D model. The reason is that in the planar model the driver is seated right above the axle while in the 3D model the driver is placed between the front and rear axle. As a result, it is subjected to a much lower vibrational impact (about 50% less).

It should be noted here that the blue curve in Fig. 16(c) actually hides four different curves representing the four wheels of the truck axle. Due to the good evenness, however, they differ scarcely. This is not the case with the next example, a road section exhibiting mediocre evenness as shown in Fig. 17. Here we can observe significant differences between the dynamic wheel loads of the truck axle in Fig. 17 with a considerable “spreading” of the blue curves (Fig. 17(c)), especially in the section between 10 and 30 m. The reason is not the longitudinal but the transversal evenness: due to considerable rutting the dual wheels of the truck axle are not loaded evenly. Instead, the wheels take an uneven share of the load as illustrated in Fig. 18.

The next three figures highlight the comparison between the 3D and the planar model (1D/2D, based on LWI). The ordinate represents the results calculated by the 3D model; the results of the planar model (LWI) are given by the abscissa. A total of 28 sections have been analyzed for this comparison each with a length of 100 m. Fig. 19 shows the comparison for the dynamic wheel loads. A close agreement can be found between both models, especially for the rms values with an $R^2 = 0.9681$ and a slope of the regression line very close to 1. This is a strong indication of a very similar parameterization of the chassis suspension (see Fig. 20).

Fig. 20 shows the comparison for the cargo load accelerations. Similar results can be found for both models with the 3D model exhibiting slightly lower accelerations (about 10% less for the peak values and 16% for the rms values). While dynamic wheel loads and cargo load...
accelerations are giving comparable results, this is not the case for the perceived driver acceleration as can be seen from Fig. 21. The acceleration in the 3D model is about half of the acceleration of the 1D model (LWI) which is particularly obvious in the case of the rms value exhibiting a regression line with a slope very close to 0.5. The reason, as explained earlier, is that in the planar model the driver is seated right above the axle while in the 3D model the driver
is placed in between the front and rear axle subjected to a much lower vibrational impact (about 50% less).

As another result of the investigations limit values in terms of target, warning and threshold values for the criteria pavement loading, cargo loading and human exposure to vibration can be established. Table 3 contains accelerations and forces which can be expected when applying the models to 3D pavement data representing motorways of good, mediocre and bad evenness. They can serve as guidelines for the proposal of limit values.

Concluding this section we can state that the 3D model for the assessment of evenness gives reasonable results comparable to existing planar models with the advantage that vibrations can be represented more realistically as could be shown in the case of the vibrational impact on the driver.

8. Conclusions

Pavements are 3D in their shape. They can be captured in three dimensions by modern road mapping equipment which allows for the assessment of pavement evenness in a more holistic way as opposed to current practice. It makes sense to use 3D vehicle models to simulate the effects of 3D surface data on certain functional criteria like pavement loading, cargo loading and driving comfort.

In order to evaluate the three criteria mentioned two vehicle models have been created: a passenger car used to assess driving comfort and a truck-semi trailer submodel used to assess pavement and cargo loading. The 3D vehicle models and their application to 3D surface data are presented to assess the pavement evenness. The results are well in line with existing single-track (planar) models. Their advantage over 1D/2D models is demonstrated by the example of driving comfort evaluation.

Existing “geometric” limit values for the assessment of longitudinal evenness in terms of the power spectral density could be used to establish corresponding limit values for the dynamic response, i.e. driving comfort, pavement- and cargo loading. The limit values are well in line with existing limit values based on planar vehicle models. They can be used as guidelines for the proposal of future limit values.
The investigations show that the use of 3D vehicle models is an appropriate and meaningful way of assessing 3D pavement evenness data gathered by modern road mapping systems.

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