The influence of suspension nonlinearities on fatigue assessment of vehicle structure

Zbyszko Klockiewicz¹, Mikołaj Spadło and Grzegorz Ślaski¹

¹Poznan University of Technology,
Maria Skłodowska-Curie Square 5, 60-965 Poznan, Poland
zbyszko.klockiewicz@put.poznan.pl

Abstract. Load spectrums for the fatigue analysis were created using suspension responses generated in a simulation of vehicle and suspension vertical dynamics nonlinear model for different conditions of vehicle use. The next stage presented was the use of finite element method and analysis of obtained stresses with its transformation to a set of cycles that are used in the determination of fatigue characteristics. The qualitative and quantitative analysis of the stresses field in the vehicle structure and suspension elements was done and later the influence of suspension responses on the fatigue assessment for most loaded parts of suspension and vehicle structure. Lastly conclusions were drawn from the results describing qualitative and quantitative influence of different road class and load conditions on fatigue assessment of vehicle structure and suspension components. Conclusion on the proposed and used methodology also was drawn.

1. Introduction
While designing the load-bearing structures of vehicles (vehicle frames, chassis) for the exploitation loads criteria, the temporary strength of the construction dependent on exploitation loads and potential dynamic overloads needs to be taken into account [1]. However it is worth noting that for normal conditions of use, the loads to which the vehicle will be subjected to fluctuate wildly, which in turn causes the fatigue strength criterion to be as important as temporary strength [2]. The widely accepted hypothesis used by the engineers for years now point to a number of factors affecting the fatigue strength, for a vehicle those factors being for example:
- the static load of the construction (responsible for the mean stress levels throughout the construction),
- dynamic loads characteristics, which in turn depend on following factors:
  - dynamic load spectrum dependent on the type of kinematic excitation, that are the result of the vehicle velocity and the irregularity profile of the road surface (for road vehicles – usually the road class),
  - the dynamic properties of vehicle’s suspension – stiffness and damping characteristics of suspension and tire;
- the stiffness of bearing structures and suspension elements.

The possibilities offered by modern CAD software allow to take into account most of the aforementioned factors, however in order to do that creating a complicated models that require high
calculating power is necessary, which causes the fatigue strength analysis process cost- and time-consuming.

The authors of this paper propose a hybrid approach which combines a relatively quick vertical dynamics simulations that allow for nonlinear suspension characteristic and implementation of diverse range of kinematic excitations, while simultaneously analyzing the influence of the structural rigidity on fatigue strength.

2. Analysis procedure
Conducting a fatigue strength analysis using the hybrid modelling method proposed in the paper required following steps to be taken:

I. creation of vertical dynamics model for the test vehicle (single axis trailer) as a quarter car model with two variants:
   a. linear,
   b. nonlinear;
   The motion ratio of suspension affecting suspension forces were already taken into account at this stage.

II. creation of a kinematic excitation generator, which could provide excitation signal for chosen road classes [3] – A, B and C in the case of this research, that could be made it inputs to the simulation model from the previous point for different vehicle speeds,

III. running a vertical dynamics simulation in which the timeseries of chosen dynamic responses are recorded to be used in further fatigue strength analysis – for the research presented in this paper sprung mass displacement was chosen,

IV. preparing the CAD model of the trailer in Siemens NX software,

V. preparing FEM model,

VI. response dynamics simulations

VII. fatigue strength calculations based on response dynamics analysis results

VIII. drawing conclusions about fatigue durability using basic stress hypothesis.

2.1. Matlab vehicle dynamics model
There were two different quarter car models used in simulations, one being linear and the other non-linear. The model (Figure 1) consists of two masses – sprung one labelled $m_s$ and unsprung one, $m_u$. Both of them have only 1 DOF – movement in the vertical direction.

![Figure 1. Quarter car model used in Matlab-Simulink.](image)

They interact with each other via spring with stiffness $k_s$ and damper with damping coefficient $c_s$, which depending on the model are either simple coefficients or (in the non-linear models) are modelled in the form of lookup tables containing spring and damper characteristics. Linear model parameters of stiffness are estimated on the base of springs characteristics in static load area and suspension motion ratios for damper and spring. Linear model parameters for damping were estimated.
by calculating the damping coefficient based on the time it took oscillations after an excitation to cease. The tire forces are modelled using stiffness $k_u$ and damping $c_u$ coefficients [4].

The characteristics of nonlinear spring and shock absorber were modified versions of characteristics acquired experimentally (described in [5]), with minor corrections made to them, to model suspension stiffness and damping instead of real spring and damper parameters. These characteristics are shown in Figure 2 and Błąd! Nie można odnaleźć źródła odwołania. All the characteristics already take into account the suspension motion ratio needed to compute the forces actually acting on suspension elements. For both front spring and damper the ratio was 0.94, for the rear spring it was 1.05 and for the rear damper 0.68.

![Nonlinear damper characteristics](image1)

**Figure 2.** Nonlinear damper characteristics.

![Suspension deflection ranges](image2)

**Figure 3.** Nonlinear spring characteristic with cumulative frequency of suspension deflections shown for C class road.

Static load of the trailer was chosen so that the suspension would enter the nonlinear working range for a sizeable portion of the time. GVW for the trailer is 750 kg, so the sprung mass value for quarter car model (which is half of the two-wheel trailer) was chosen as 350 kg.
The linear model was calibrated so that it estimated the almost linear part of the nonlinear characteristic. There were two main nonlinearities in the other characteristic – first one was that because of preexisting load, the spring force starts at around -400 N for suspension deflections just over 0 mm. After that almost immediately the linear working range of a spring is entered (for which the stiffness coefficient is roughly 22 kN/m), which ends at 150 mm of deflection, at which point the bump stop starts being compressed. This causes the spring force to rise significantly, almost doubling the stiffness coefficient. At 220 mm of spring deflection the coils of the spring come into contact – further compression would require the material or its supports to deform, which requires significantly greater forces, which can be seen as an almost vertical line at the end of the characteristic.

2.2. Kinematic excitation generation
Kinematic excitations were modelled using procedures based on [6], which discusses the generation of road profiles according to classes defined in ISO 8608 standard. To obtain kinematic excitation from road profile, the use of tire model is normally needed, in the presented case the tire model was replaced by defining the shortest existing profile wavelength to be 0.2 m, which is the typical passenger car tire contact length.

The road irregularity profiles used in simulations belonged to classes A to C, as defined in ISO 8608 standard. The road profiles were generated using method described in [7]. The Matlab function that generated profiles used geometric mean values of the vertical displacement power spectral density for each road class as they are given in ISO 8608 [6]. Road waviness was constant and set to 2. The vertical displacement PSD was then calculated for a range of angular spatial frequencies with its borders defined by set longest (90 m) and shortest (0.2 m) wavelengths. These wavelengths were chosen as the tire filters unevennesses shorter than 0.2 m, as that is the average contact length of tire-pavement pair for passenger cars and passenger car trailer tires. Unevennesses longer than 90 m would require a very high speed of around 180 km/h to be noticeable, which is rarely accomplished on public roads due to speed limits and safety reasons. Randomly distributed phase shift was also calculated and both of these were later used in an equation that created a number of profiles for every frequency, that were ultimately summed together to create road profile from a chosen road class.

3. Results of vehicle vertical dynamics simulations

3.1. Simulation tests cases and model parameters
Two sets of profiles far all classes were generated, so that one of them could be used as a surface under right and left wheel. Once the kinematic excitation profiles were generated, they were used as an input to a series of simulations. The main output of the simulation was the displacement of sprung mass, which was later used as an input to a fatigue strength simulation in NX Siemens. Twelve such signals were registered, as there were two profiles for each of the three road classes, and all of them also had a linear and nonlinear variant.

| Table 1. Parameters of the linear quarter car model used to generate inputs to FEM analysis. |
| Sprung mass for curb weight + load | $m_s$ | 700 kg (350 for quarter-car) |
| Unsprung mass | $m_u$ | 20 kg |
| Tire stiffness | $k_u$ | 175 kN/m |
| Tire damping coefficient | $c_u$ | 500 Ns/m |
| Suspension linear spring stiffness | $k_s$ | 17.5 kN/m |
| Suspension linear damping coefficient | $c_s$ | 1000 Ns/m |
The vehicle’s speed was dependent on the road class for which the simulation was ran. For the A class road it was 108 km/h, B class 80 km/h and for the C class it was 72 km/h. Other parameters of the test vehicle are given in Table 1.

### 3.2. Simulation tests results
Simulation tests results were presented in further figures comparing results obtained for linear and nonlinear suspension models.

**Comparison between linear and nonlinear models - sprung mass displacement, road class A**

**Comparison between linear and nonlinear models - sprung mass displ. cumulative distributions**

**Comparison between linear and nonlinear models - suspension force, road class A**

**Comparison between linear and nonlinear models - suspension force cumulative distributions**

*Figure 4.* Comparisons for sprung mass displacement and total suspension forces for road class A.
The figures for class A show slight differences between linear and nonlinear model. Sprung mass displacement values show similar values for both models, which can be also observed in the timeseries. Force values seem to change more rapidly for the nonlinear model, and the narrower range of forces can be seen for a linear model, which is to be expected, as the same excitations might cause the suspension to enter nonlinear range, where the forces grow much more rapidly.

Figure 5. Comparisons for sprung mass displacement and total suspension forces for road class B.

For the class B road similar trends can be observed as for the class A one – the suspension force graph for the nonlinear model has more sudden spikes in value (presumably where suspension enters nonlinear range for spring) and slightly narrower range for sprung mass displacements, while the range for suspension forces is wider. The difference between models is better visible, as could be
expected from a worse quality road, where the suspension is more likely to enter nonlinear working range.

Figure 6. Comparisons for sprung mass displacement and total suspension forces for road class C.

C class road once again shows similar distribution of both forces and displacements – however the range for displacements seems to be more equal between linear and nonlinear model. Extreme force values for the nonlinear model were greater by roughly 500 N, compared to 200 N for the B class and 100 N for the A class.

Table 2 contains maximum and minimum values of suspension deflection for all three road classes for both linear and nonlinear models, as well as the percentage of deflections that entered the nonlinear
range of suspension spring (red line in Figure 3). The absolute values between classes A and B do not differ significantly, more noticeable difference can be seen for C class road. When it comes to percentage values, in every case the value for nonlinear model is much higher than for the linear one on corresponding road class, being more comparable with the linear value for the road of the next class (for example 3.6% for A class, nonlinear and 5.24% for B class, linear).

| Susp. deflection | A class road | B class road | C class road |
|------------------|--------------|--------------|--------------|
|                  | Linear       | Nonlinear    | Linear       | Nonlinear    | Linear       | Nonlinear    |
| max. value [m]   | -0.1284      | -0.1390      | -0.1203      | -0.1269      | -0.0986      | -0.1083      |
| min. value [m]   | -0.1478      | -0.1520      | -0.1549      | -0.1570      | -0.1752      | -0.1731      |
| Percentage in nonlinear range [%] | 0.00 | 3.60 | 5.24 | 14.90 | 20.1 | 37.97 |

Figure 7. Comparison of sprung mass acceleration distributions for different road classes and different suspension models.

The graphs show differences between sprung mass acceleration distributions for different road classes and vehicle models. The differences are most clearly visible for the C class road, which has significantly broader range of registered accelerations, while classes A and B are somewhat similar for both linear and nonlinear models. The other trend that can be seen is that for all three road classes the range for nonlinear models is broader than for linear ones, especially when it comes to positive accelerations – for C class for example the extreme value for linear model is around 3 m/s^2, while for nonlinear it is closer to 6 m/s^2, while for negative accelerations the values are 4.1 m/s^2 and 4.6 m/s^2, respectively. Similar relations can be seen for other road classes as well.

4. Results of FEM simulation for fatigue strength estimation

4.1. Method and tools used for fatigue strength analysis

Using the sprung mass displacement data acquired from Matlab-Simulink model for both linear and nonlinear models, fatigue strength analysis was carried out. The goal was to establish the influence of
nonlinearities in the suspension characteristics on the durability of the vehicle’s supporting structure (trailer frame) depending on the road class, on which the vehicle travels. Comparative analysis was carried out for road classes A, B and C.

In order to estimate fatigue strength of the structure, the following steps were taken:

1. In the first phase the supporting structure’s dynamic response analysis was carried out using the finite element method. Solver 103 from NX Siemens “Response Dynamics” software was used for the task, which uses modal superposition method for its calculations. Those analysis involved the implementation of sprung mass displacements sequences, registered beforehand separately for the left and right side of the trailer. Additionally the drawbar had its translations in all three axis blocked, while the rotations were allowed in the place where ball joint is located. The calculations concluded with stress values obtained for every finite element in the structure.

2. In the second phase durability is assessed with the aptly named “Durability” module in the same software. The method chosen was stress life analysis, which is suitable for high cycle fatigue. The maximum stress level is generally very low and the number of cycles to failure is generally very high. Durability uses maximum principal stress amplitudes and corresponding mean stresses for stress life equations. Mean stress effects in the damage calculation were included, and the stress life equation of the S-N curve was written using Goodman’s method [8]:

\[
\frac{\Delta \sigma_1}{2} = \left( S_u - \sigma_m \right) \frac{\sigma'_f}{S_u} (2N_f)^b
\]

where:
- \( \Delta \sigma_1 \) - the maximum principal stress amplitude,
- \( 2N_f \) - the number of reversals to failure,
- \( \sigma'_f \) - the fatigue strength coefficient material property,
- \( b \) - the fatigue strength exponent material property,
- \( \sigma_m \) - the mean stress of the cycle along the principal axis,
- \( S_u \) - the ultimate tensile strength material property.

The software uses Newton’s method to solve for the reversals to failure by a given stress amplitude. Furthermore in the applied method the schematization of random loads is done using the Rainflow method, while the damage accumulation in the material is described by the Palmgren-Miner hypothesis [9]. After the stress or strain amplitude and mean stress are determined for each cycle, the number of cycles to failure are determined using the selected life criterion (S-N curve). The damage of the cycle was then calculated by:

\[
D_i = \frac{1}{N_f}
\]

where \( D_i \) is the damage caused by the \( i \)-th cycle and \( N_f \) is the number of cycles to failure for the \( i \)-th cycle. Applying the Palmgren-Miner rule, the total damage, \( D_T \), of an event was calculated by summing up the damage caused by each cycle:

\[
D_T = \sum_i D_i
\]

The number of times the strain history can be repeated until failure occurs, \( N_{R_f} \), is the inverse of the total damage:

\[
N_{R_f} = \frac{1}{D_T}
\]
4.2. Results of fatigue strength analysis

The aforementioned methods are basic and widespread standard for fatigue strength analysis in the preliminary stages [10]. Using them it was possible to calculate the number of cycles before fatigue damage appears for every finite element and analyzed load case:

- various road excitations - road classes A, B and C,
- input signals of sprung mass movement from dynamic simulation of two types of suspension – linear and nonlinear.

The results of the analyses are shown in Figure 8 to Figure 13, which show the number of 10-year long exploitation periods (it was assumed to be 600 hours per year). This means for example that the value of 40 for a given finite element tells that this element can work for $40 \cdot 10 = 400$ years with low risk of fatigue damage appearing, for a given road class.

In order to quantitatively compare simulation effects, the number of 10 year exploitation cycles were compared for two chosen nods, for which the registered stresses were high. It is worth noting that for all cases the exactly same finite elements were compared to ensure its reliability. The chosen nods are shown in Figure 12.

![Figure 8](image_url)

**Figure 8.** The amount of 10 year load cycles for excitations from an A-class road for a linear suspension model.
Figure 9. The amount of 10 year load cycles for excitations from an A-class road for a nonlinear suspension model.

Figure 10. The amount of 10 year load cycles for excitations from an B-class road for a linear suspension model.

Figure 11. The amount of 10 year load cycles for excitations from an B-class road for a nonlinear suspension model.
Figure 12. The amount of 10 year load cycles for excitations from an C-class road for a linear suspension model.

Figure 13. The amount of 10 year load cycles for excitations from an C-class road for a nonlinear suspension model.
Based on the figures’ analysis the conclusion can be drawn, that the risk of fatigue cracks forming is low for both linear and nonlinear suspension model in the entire assumed exploitation period. This probability is much higher for road classes B and, especially, C, for both linear and nonlinear versions.

The influence of the suspension type can be most clearly seen for the C-class road, with diminishing influence as the road class becomes smoother – the difference is the lowest for A-class road, while being in between A and C for the B-class road. The reason for that is that the worse the surface quality is, the more often suspension will enter the nonlinear working range, which generates significantly greater forces on the structure.

Table 3 contains the results of the number of 10 year exploitation cycles for the aforementioned, chosen two nods. The effects of the implementation of nonlinear spring characteristic can be clearly seen for all road classes. Additionally the ratio between linear and nonlinear number of cycles was calculated:

\[ \text{ratio} = \frac{\text{number of cycles linear}}{\text{number of cycles nonlinear}} \] (5)

Differences in the number of cycles between linear and nonlinear cases range from around 5 to over 100 times less cycles until fatigue damage appears for nonlinear model.

Table 3. The results of fatigue strength simulation for all analysed road classes.

| Element no | A class road linear | A class road nonlinear | B class road linear | B class road nonlinear | C class road linear | C class road nonlinear |
|------------|---------------------|------------------------|---------------------|------------------------|---------------------|------------------------|
| 17198      | 3.9 \cdot 10^{11}   | 0.68 \cdot 10^{11}    | 35432               | 4320                   | 260                 | 2.2                    |
| 29125      | 11 \cdot 10^{11}    | 1.8 \cdot 10^{11}     | 8417                | 740                    | 104                 | 0.95                   |

5. Conclusion
The article present the results of research that merges the suspension dynamics analysis for random kinematic excitation from road irregularities of different classes with fatigue strength analysis of a single axis trailer bearing structure. Thanks to the use of presented hybrid fatigue strength analysis procedure it is possible to relatively quickly achieve results concerning fatigue durability of the whole structure. The use of sprung mass displacements as inputs to FEM simulation allows to estimate stress values, their fluctuations and, as an end result, also the fatigue strength durability for the load bearing structure of the trailer.

The research was conducted simultaneously for linear and nonlinear suspension models, which showed that the implementation of a more complex nonlinear model is important, as it affects the resulting suspension force both from the suspension spring, as well as the damper. The exact value of work in the nonlinear range varied for different road classes, being the lowest for A class road (3.8%), higher for B class road (around 15%) and reaching almost 38% for the C class road.

Entering the nonlinear working range of a suspension spring’s characteristic causes the accelerations that act upon sprung mass to rise significantly, which in turn increases inertial forces in the bearing structure. This causes the fatigue durability to drop. The influence of nonlinearity on the number of cycles before fatigue damage becomes more pronounced for road of a worse quality, but it
is also noticeable for good quality road (like A class road), because of the presence of damper nonlinearity.

For all road classes similar trends can be observed for both sprung mass displacements and total suspension forces. The range for sprung mass displacements is narrower for linear model, while the opposite is true for suspension forces. This is to be expected – same kinematic excitations cause greater forces in the suspension when model enters nonlinear range, but at the same time the damping forces are also greater, limiting sprung mass movements. The difference in displacements is smaller than the one in forces, which can also be observed in the timeseries for all road classes – the one for nonlinear model changes value more rapidly and many spikes in value can be observed.

The obtained results concerning fatigue durability as a number of cycles before fatigue damage are based on the basic model of durability assessment. There are more complex models that allow to account for other factors that affect fatigue strength, that is why the obtained results when it comes to the absolute value might include some errors due to the method chosen, however they allow to compare the estimated durability between analyzed cases of exploitation loads. They also prove the influence of the model characteristics nonlinearity on the fatigue durability of the load bearing structure of the trailer. The obtained results indicate that the risk of fatigue damage for road classes A and B is not great, while the constant use on the C class road poses such risk in the span of 10 years of exploitation. This proves that considering fatigue durability is a must during the designing process, while also showing how challenging it is to account for all possible exploitation loads and conduct a proper analysis process, in which nonlinear suspension models need to be included, especially if the vehicle is expected to operate around its GVW.

At the same time, the obtained results suggest that using linear characteristics of suspension elements bears the risk of overestimating fatigue durability by the factor of a few to few dozen times in comparison with the results for the nonlinear model.

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