FATIGUE LIFE CALCULATION FOR MILITARY VEHICLES UNDER REAL SERVICE CONDITIONS

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THEME

Failure Prediction and Assessment

KEYWORDS

Fatigue Life; Off-Road Vehicles

SUMMARY

The development of a new military vehicle needed a light weight option because these vehicles have to be transported by aeroplane. The optimisation of the structure has to be done using advanced methods for the design including fatigue life calculation. In this paper it is shown how one of the most loaded components – the wheel hub- was designed. A combination of outdoor tests, indoor tests and simulation was done. In the first step the use of the vehicle on varying terrain was investigated together with the customer. This was necessary for the design target.

In the next step simulations using ADAMS and measurements with a prototype of the vehicle supported with measurement rims on all wheels were done. It was possible to determine the loading histories of the forces on all interesting road classes. Based on these data a test-rig procedure was established to drive the loading history in the laboratory.

Based on the measured forces in the wheel, calculations using FEM and fatigue life software were done. Before test-rig results or vehicle results were available it was obvious from the fatigue life calculation that failure would be expected very early on in the notch in the wheel hub. Because it was proved possible to get reliable fatigue life data from calculations, the fatigue life calculation is now a permanent part of the development process.

FATIGUE LIFE CALCULATION FOR MILITARY VEHICLES

1: Introduction

For the development of “Fennek” it was an important aim that the vehicle should be as light as possible. To ensure that the vehicle was suitable for the required conditions, these were taken into consideration as important ancillary conditions for the calculation.

The average performance can be anticipated to be made up as follows:

25 % roads
40 % track roads
20 % light terrain
10 % semi-difficult terrain
5 % difficult terrain

Figure 1: Military vehicle Fennek (weight 10.8 tons)

At the beginning of the development phase we were aware that very heavy demands were going to be made on the chassis in particular. We therefore carried out extensive simulations with FEM (NASTRAN), MKS (ADAMS) and fatigue life calculation programs (winLIFE). These were a help in the early stages of the construction phase. The fatigue life calculation was carried out based, among other things, on the three forces and 3 moments affecting the wheel. These were measured with a test rim whereby the rotating wheel was modeled by static equivalent loading cases which leads to a multiaxial loading.

2: Measurement of the real conditions of use

Once the proto-type was available, we carried out extensive measurements of the forces affecting the wheels. These were done on test ground (figure 2). For these tests the vehicle was fitted out with GPS sensors, video measuring technical appliances, and acceleration sensors and, most importantly, test wheel-rims so that a detailed documentation of the driven route could be made. In this way it was possible to identify the various types of road surface driven on and to allocate these to the required type of route. In order to match the driven route to the individual road surface categories as in [1, 2], the measured unevenness was recorded according to the vertical movements of the wheel as follows: The measured vertical movements of the test rim were used to calculate the road surface unevenness (figure 3). The effective values and spectral density of the road surface unevenness could then be ascertained from these calculations. Figures 3 and 4 show two cobbled street routes which are actually identical. Because of the wear, however, the old route shows smaller effective values than the new route. Using the transfer function of the right rear wheel of the suspension (figure 5) the unevenness of the road was calculated from the measured wheel load.

The following diagrams show how all the driven routes are graded and arranged in the diagram according to Braun [1]. This enables the measured road profile to be matched to the type of route without any doubt which is very important for the current development as well as for the final evaluation (figure 6).

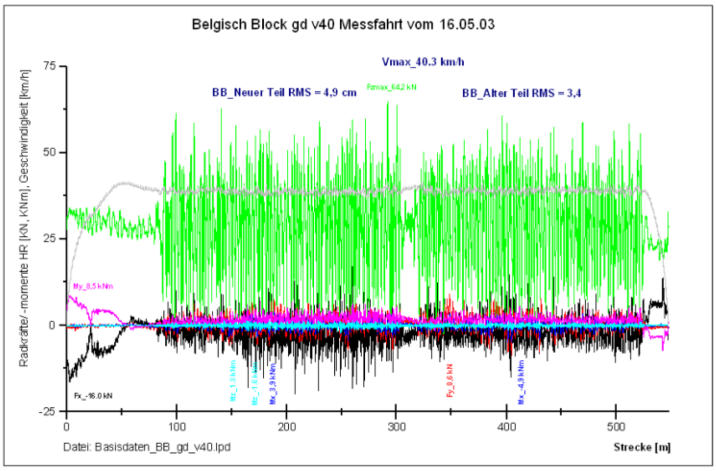

Bild 1: BB_gd_V40 Radkräfte und -momente HR, Fahrtrichtung Norden nach Süden

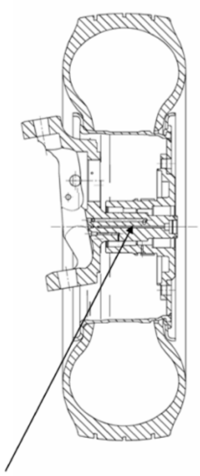

Figure 2: Wheel forces and moments measured by the equipment on the right

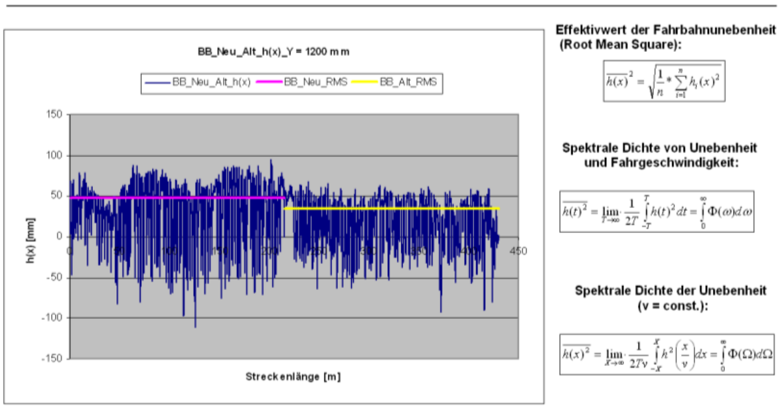

Bild 2: Unebenheitshöhe über Strecke: BB, neu alt Spur rechts, RMS

Figure 3: **Unevenness and effective value of the cobbled street route, old and new**

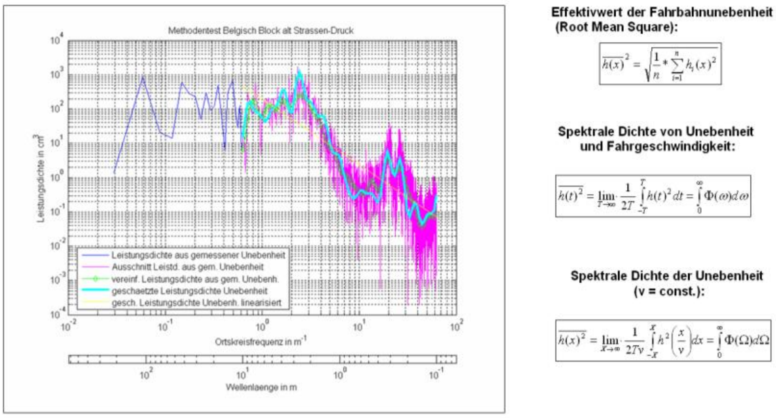

Bild 3 Leistungsdichtespektrum $\Phi_h(\Omega)$ BB Alter Teil (Messung - Rücktransformation)

Figure 4: **Spectral density of the unevenness ascertained from the data from figure 3**

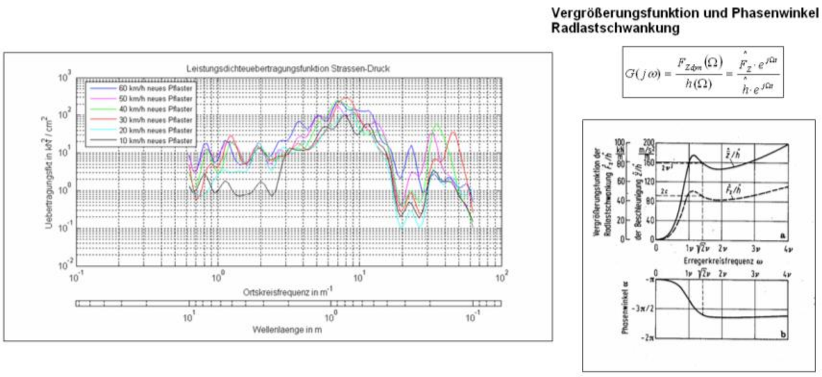

Bild 4 Übertragungsfunktion BB, Neuer Teil | G(jΩ) |² Strassendruck

$$|G(j\Omega)|^2 = \frac{\Phi_{Fzdyn}(\Omega)}{\Phi_h(\Omega)}$$

Figure 5: **Transfer function cobbled street, new part**

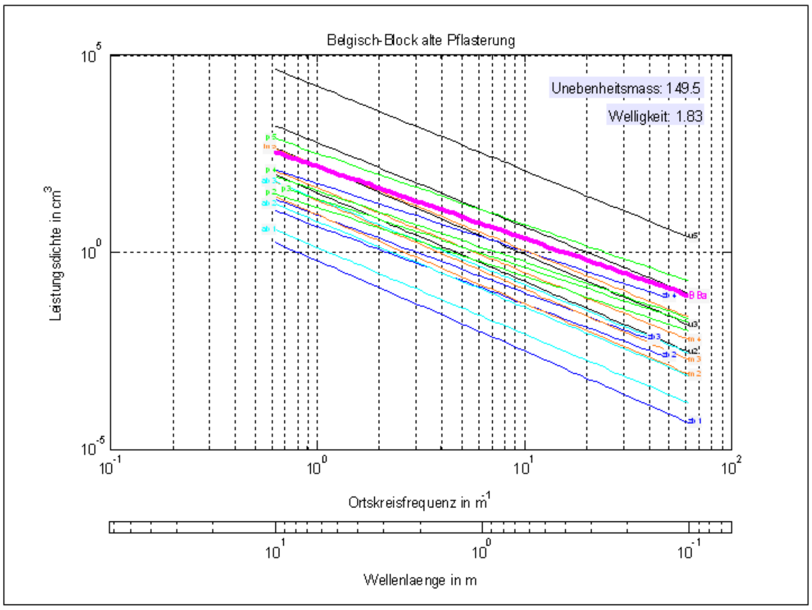

Figure 6: Spectral performance density and sorting according to Braun [1]

3: Combination of FEM calculation with Fatigue Life Analysis

The Fennek wheel-hub is a component subjected to particularly high demands (figure 7) and here it is explained how the fatigue life calculation was carried out.

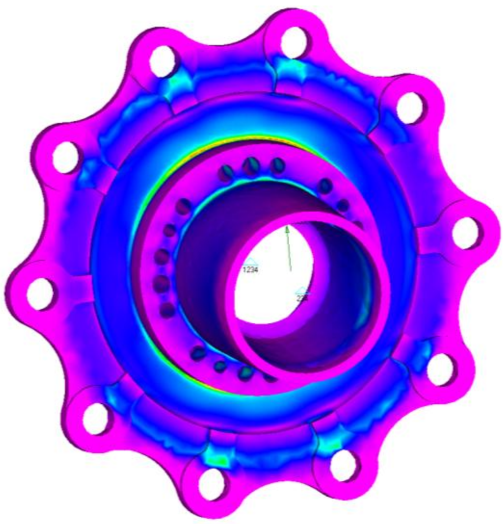

Figure 7: FE-Model of the Fennek wheel hub

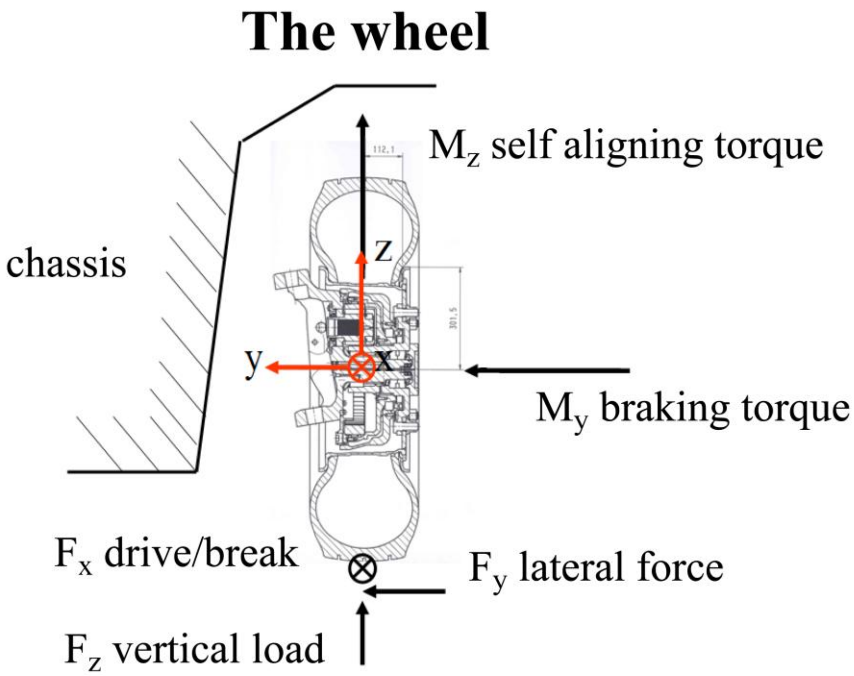

Figure 8: Fennek wheel showing the built-in wheel hub and the forces acting on it

All the sizes affecting the wheel, the three forces and three moments (figure 8), were recorded using a test-rim and were available with a high sampling rate for each type of route separately.

To carry out the fatigue life calculation of the wheel hub with the aid of static FEM-calculations, the wheel hub, which in reality rotates, was reduced to a standing wheel hub affected by rotating forces. In order to take into account all the sizes (forces and moments) which in reality have an influence during the rotation, the unit load cases are defined for each angle interval. The size of the angle interval was 30 degrees which has been shown to be sufficient. Since this unit load case is only valid if the wheel carrier is in this area, the axle load is divided between the number of unit loads according to the number of load curves. These are only not equal to zero if the equivalent load case is operative. In this way the rotation of the wheel carrier can be calculated by superimposing these static load cases (figure 9, 10, 11). The procedure how to carry out for the vertical load has to be performed for each active loading. This leads to the load cases shown in figure 12.

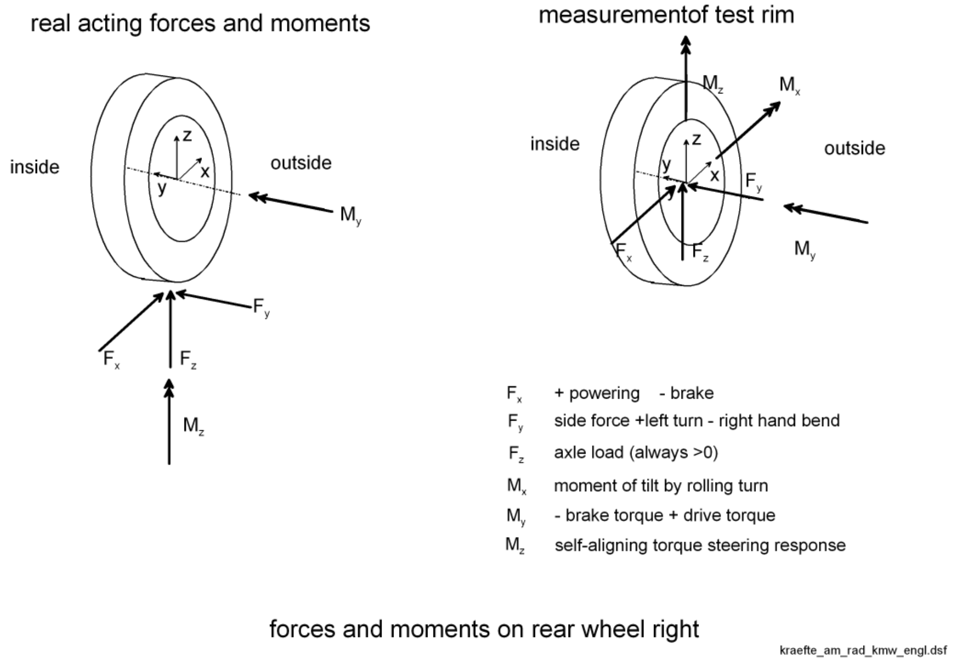

Figure 9: Forces and moments on the wheel and the corresponding measured sizes and test-rim

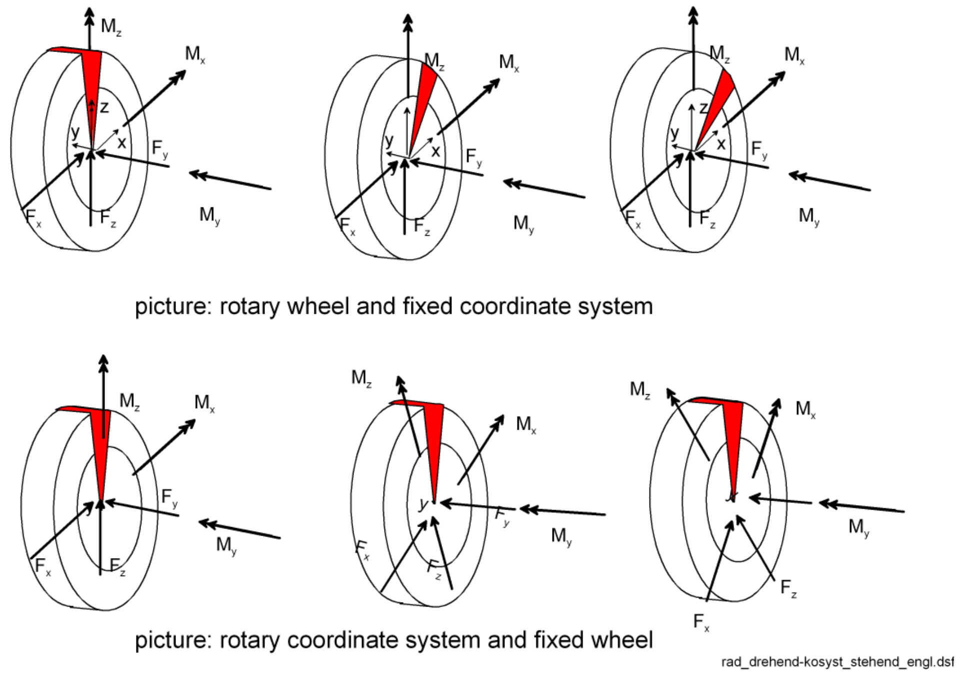

Figure 10: Real system: rotating wheel hub with corresponding fixed coordinate system (above) and alternative system (below) with fixed wheel and rotating forces

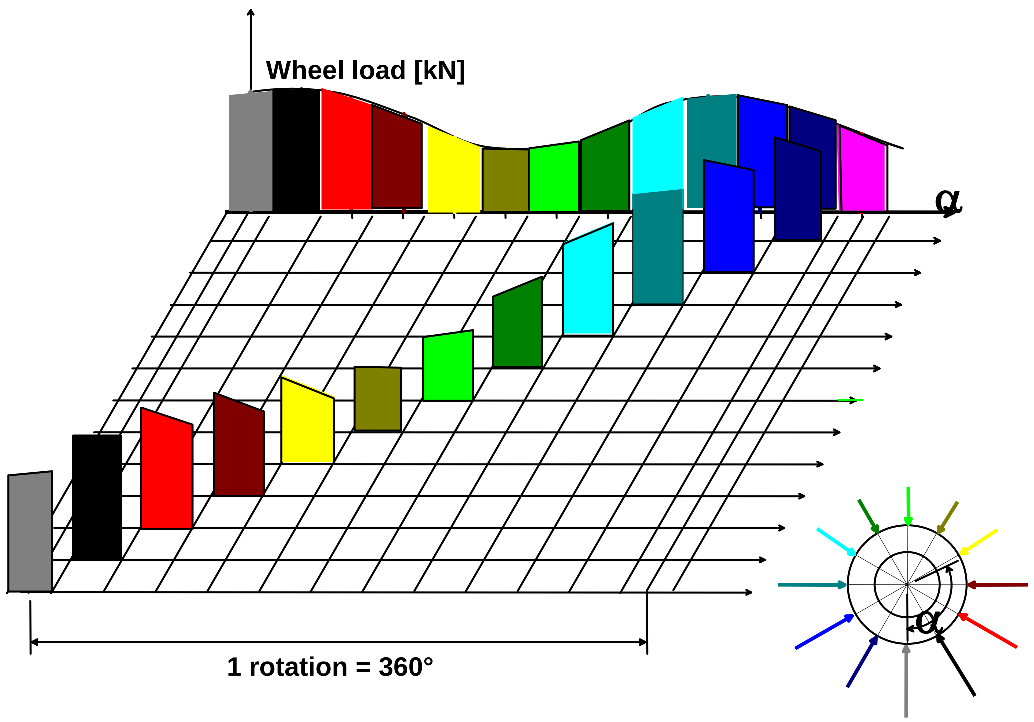

Figure 11: Measured wheel load (above) and equivalent 12 loadings which are non zero only in an angle range together with the unit load case

Load size	Number of unit load cases	Angle difference	Channel number
Fx	12	30	1-12
Fy	1	360	13
Fz	12	30	14-25
Mx	12	30	26-37
My	1	360	38
Mz	12	30	39-50
Total	50		

Figure 12: Definition of equivalent unit load cases with angle windows

4: Getting fatigue life curves for the calculation

To predict fatigue life estimation the elastic stress method and the local strain approach can be used. For the stress based concepts this was done using the approach

- according to Hück, Trainer Schütz [3, 4]
- according to FKM [4]

for creating S-N curves for the critical point in the notch.

For the strain life (local strain approach) concept the

Uniform Material Law according to Boller, Seeger [4]

and

measured data from a data base

were used. Because we have a high cycle fatigue problem the stress based method was preferred. The strain life method was carried out only for comparison.

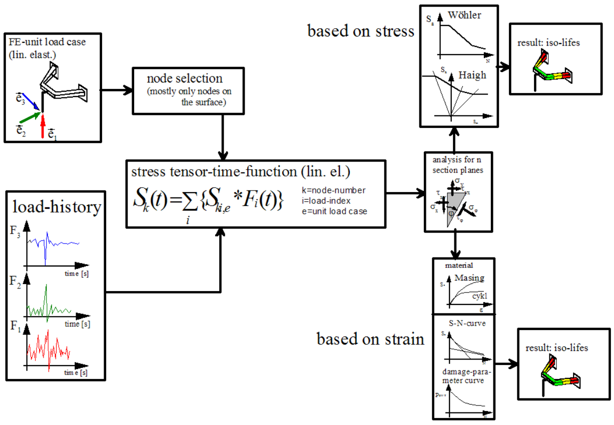

Figure 13: Schematic view of the procedures for fatigue life estimation

5: Results

The wheel hub has a distinct notch where the cracks start. The stresses/strains lie in the elastic range and there is a problem with higher stress reversal numbers (HCF) where the loading is mainly below the endurance limit. The stress based S-N curve calculation method was used and the range below the endurance limit was calculated with the fundamental Miner-modification. The S-N curve was modified with the aid of the stress gradient in the notch [5] to consider the stress related material support. To understand how the damage occurs on the individual routes, the equivalent stress history was shown together with the road profiles (figures 14 and 15).

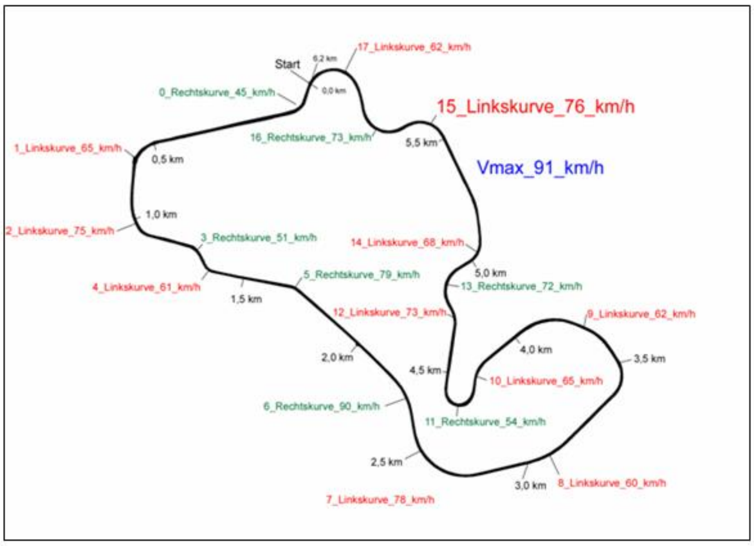

Figure 14: : Circuit route Grüneberg, route plan

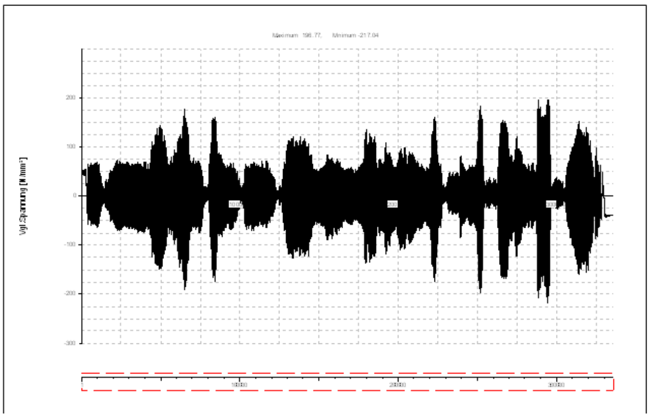

Figure 15: : Circuit route Grüneberg, equivalent stress history

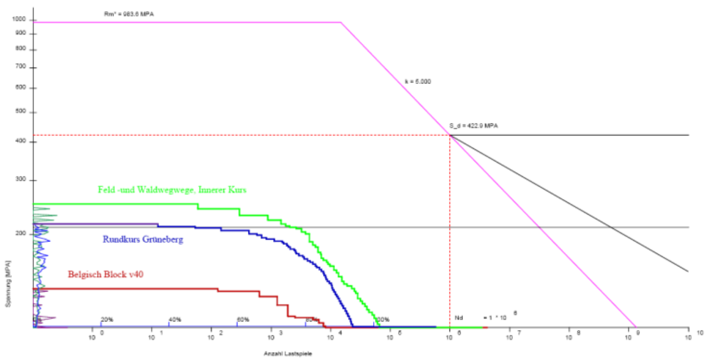

Figure 16: : Stress S-N curve together with amplitude collectives and percent of damage

Figure 16 shows the stress amplitude spectrum on different routes together with the S-N curve of the critical point. Because of the static weight allocation, the vehicle has a higher loading on the rear axle. Since the components of the front and rear axles are identical, the rear axle has a shorter fatigue life due to the greater wheel load. Figure 17 shows the results of the fatigue life calculation as a percent of the route length. The fatigue life of the rear wheel on the cobbled road route was therefore set at 100%. As expected, the wheel hub of the rear wheel has a much lower fatigue life. Furthermore, the chart shows what a large influence the type of road has. The damage is greatest on the course FWI (field –woodland –inside) while on the course RK (round route) the least damage occurs.

Road Type	Front Wheel right	Rear wheel right
BB	192	100
FWI	156	59
RK	63200	780

Figure 17: Percent of the route driven before failure

6: Conclusion

We were able to predict an area of failure and insufficient fatigue life at an early stage of the construction – before any prototype was available. Despite these predictions, the prototype was made, and we saw that the calculated damages actually occurred.

Changes in the geometry (notch) were made followed by a successful new design.

The influence of different surfaces on the fatigue life was estimated (shot blasting, rolling) to show how the existing “badly designed” parts could be improved.

The influence of driving cycles on the fatigue life was quantified; different use-scenarios could be estimated

In combination with experimental results (sufficient statistic basis) the real fatigue life can be estimated. Used in this way, fatigue life calculations are helpful in speeding up the development and are available to accompany all future developments.

REFERENCES

[1] BRAUN - Untersuchungen von Fahrbahnunebenheiten und Anwendungen der Ergebnisse, Dissertation TU Braunschweig 1969

[2] MITSCHKE - Dynamik der Kraftfahrzeuge, Band B Schwingungen Kapitel 5 Stochastische Unebenheiten, Springer Verlag 3. Auflage 1987

[3] HAIBACH - Betriebsfestigkeit, VDI-Verlag; Düsseldorf 1989

[4] HAIBACH, E, BERGER, C, HÄNEL, B, WIRTHGEN, G, ZENNER, H, SEEGER, T - Rechnerischer Festigkeitsnachweis für Maschinenbauteile, Forschungskuratorium Maschinenbau, Lyonerstr. 18, Frankfurt/M., Heft Nr. 183-1, 1994

[5] HÜCK, M, THRAINER, L, SHCÜTZ, W - Berechnung von Wöhlerlinien für Bauteile aus Stahl, Stahlguß und Grauguß, Synthetische Wöhlerlinien, Verein deutscher Eisenhüttenleute Arbeitsgemeinschaft Betriebsfestigkeit, 1981

[6] BOLLER, C, SEEGER, T - Materials Data for Cyclic Loading, Part A: Unalloyed Steels, Elsevier Science Publishers B.V.1987, ISBN 0-444-42870-4

[7] KMW - Fahrversuche am Fennek, internal Report

[8] GUDEHUS, ZENNER - Leitfaden für eine Betriebsfestigkeitsberechnung Kapitel 5 Stahl und Eisen, 4. Auflage, Verlag Stahl Eisen, Düsseldorf, 1999

[9] Steinbeis Transfer Centre: Manual to the software winLIFE, 2008

[10] HÄCKH, J, WILLMERDING, G, KLEY, M, BINZ, H, KÖRNER, T - Rechnerische Lebensdauerabschätzung von Getriebegehäusen unter Einbeziehung realer multiaxialer Belastungen, DVM-Tagung Fulda vom 5. bis 6.6. 2002, VDI-Berichte N2. 1689, Seite 303 - 317 2002

[11] KÖRNER, T, DEPPING, H, HÄCKH, J, WILLMERDING, G, KLOS, W - Rechnerische Lebensdauerabschätzung unter Berücksichtigung realer Belastungskollektive für die Hauptwelle eines Nutzfahrzeuggetriebes, DVM-Tagung Fulda vom 5. bis 6.6. 2002, VDI-Berichte N2. 1689, 2002 Seite 275 - 285