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DYNAMIC SIMULATION AND ESTIMATION OF STRESS SPECTRA FOR RAILWAY COMPONENTS

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Abstract

An investigation method of fatigue life of railway components is presented and applied to the study of the axle of a railway locomotive.
ADAMS/RAIL was used to realise a detailed model of the ADtranz E412 railway locomotive.
The results of the ADAMS simulations, covering 3 km on irregular track, were used as input for the calculation of the load spectrum. The *rainflow cycle counting method* was employed. By means of statistical approach, it was possible to build the load spectrum relative to 300.000 km, and to estimate failure probability on the axle.
The results obtained with the numerical model are quite satisfying and similar to measured data, though passage over switches and cornering should be simulated to obtain significant failure probabilities.

KEYWORDS: railway vehicles, dynamic analysis, load spectra, fatigue damage

1. INTRODUCTION

The correct dimensioning of a railway component is determined through the laboratory tests imposed by the International Regulations [1]. The limit of this procedure is connected to the load spectrum established by the Regulations, which is very far from the real spectrum acting on the component, due to the effective working conditions. So, even if necessary for the homologation, Regulation's tests are not very significant in predicting the real failure probability of the component. This leads many constructors to research the effective load spectra through experimental measurements [2], in order to estimate the fatigue–life of a component with a better precision. The costs relative to this operation have driven, in the last few years, to employ numerical simulations of working conditions, using multibody codes, to determine the real load spectrum.

This is the starting point of the method purposed, where the basic idea is to find the real load spectra by using ADAMS simulations. A scheme of the method is reported in Figure 1.

The procedure can be summarised in the following points:

1- realisation of the model with ADAMS/RAIL;
2- numerical simulation of most significant working conditions (irregular track, traction, breaking, curve, switch);
3- construction of the load spectrum on the component starting from the forces acting on it
4- construction of the S-N curve relative to the component ;
5- determination of the failure probability.

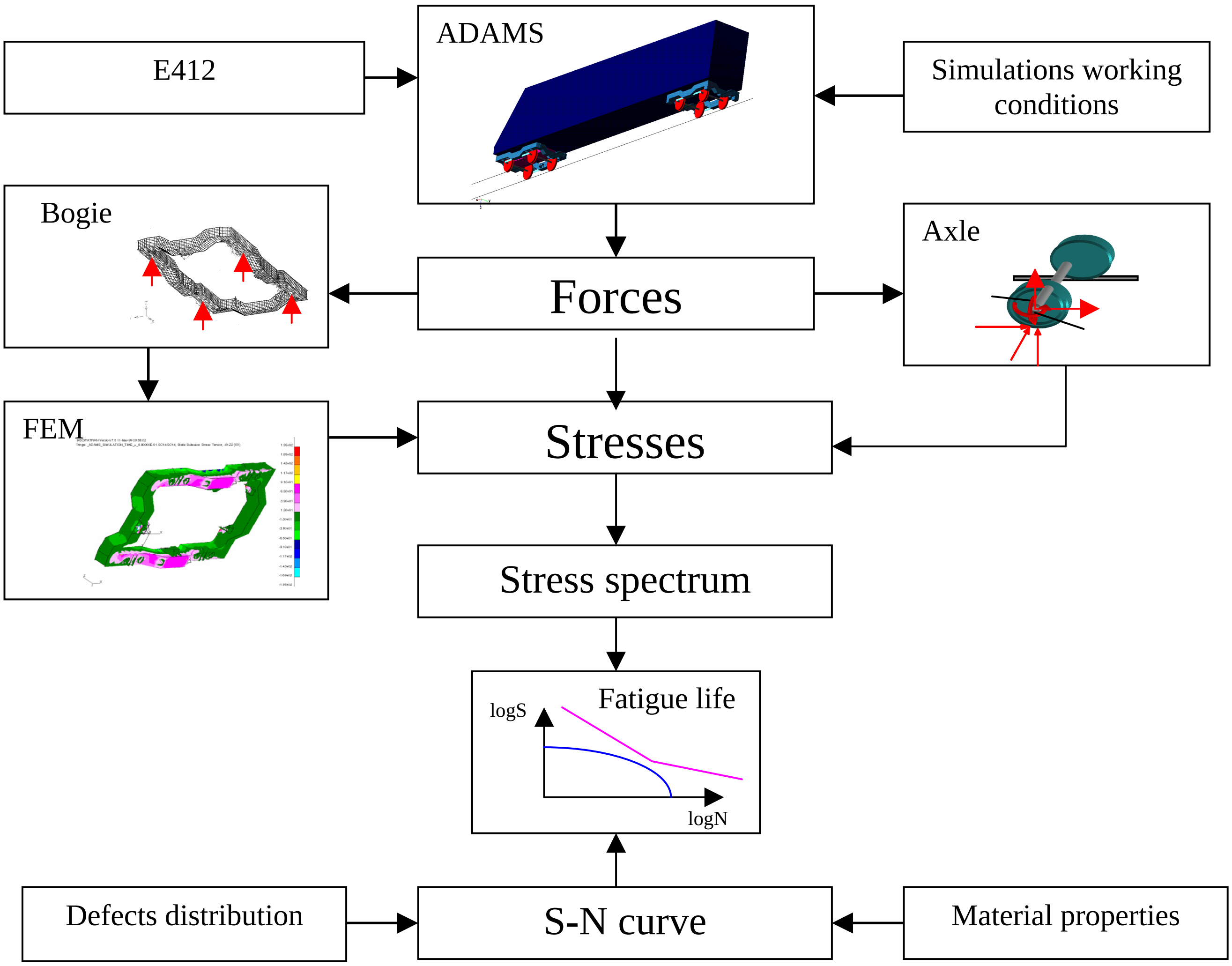


Figure 1 Scheme of the method.

2. DYNAMIC SIMULATIONS

2.1 DESCRIPTION OF THE MODEL

A scheme of the ADAMS model of ADtranz E412 vehicle is reported in Figure 2.

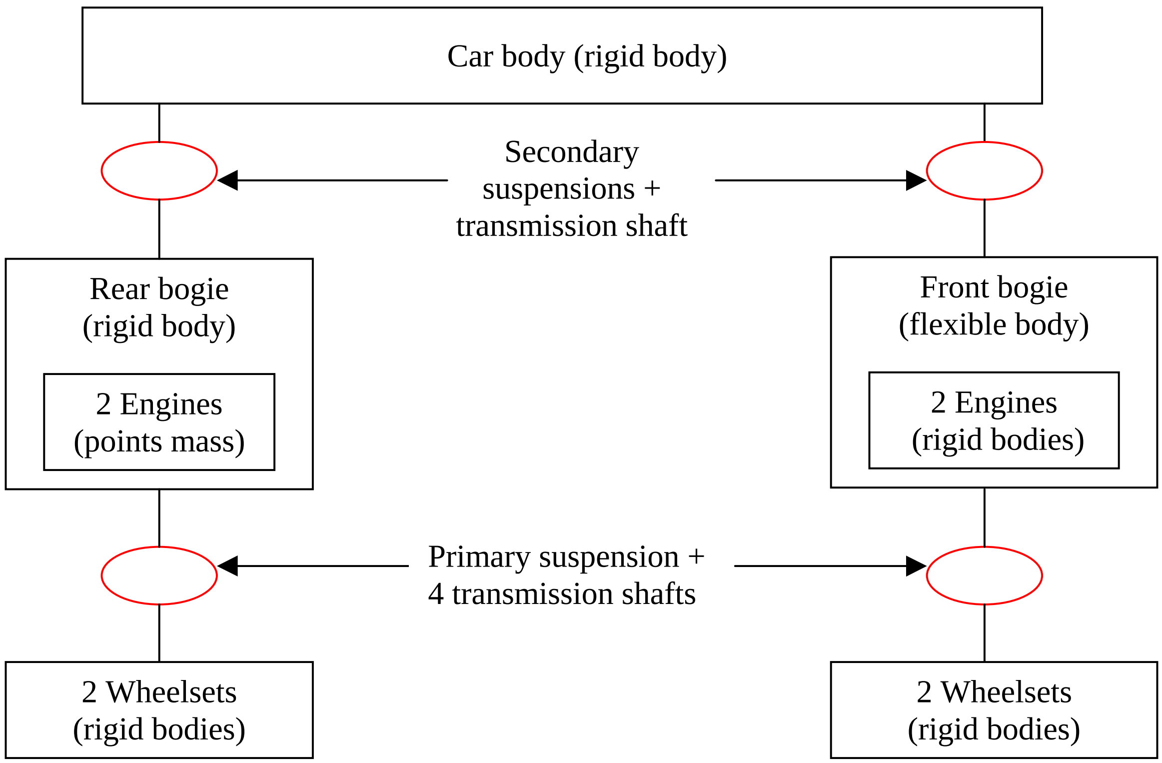


Figure 2 Scheme of the model.

ADAMS' model has 198 d.o.f. and presents a particular accuracy in the front part where:
- the bogie was introduced as a flexible body;
- the engines were reproduced as rigid bodies, with their connections to the bogie and to the car body.

The front bogie has 21 hardpoints for the connections to the primary and secondary suspensions and to the transmission devices. The information about the modal deformation of the bogie was obtained from a FE model of the bogie, realised with the NASTRAN code by ADtranz. The choice of giving more precision only to the front part of the model is due to the need of reducing the simulations' time.

The front bogie is the only flexible body introduced in the model: all the other components (car body, rear bogie, wheelsets, transmission devices) were considered as rigid bodies. The presence of the engines in the rear part of the model is reduced to 5 points mass.

All the spring-dumper elements were introduced as *bushing-elements*, so non-linearity has not been taken into account. Figure 3 shows the front part of the model (the visibility of the engines has been removed).

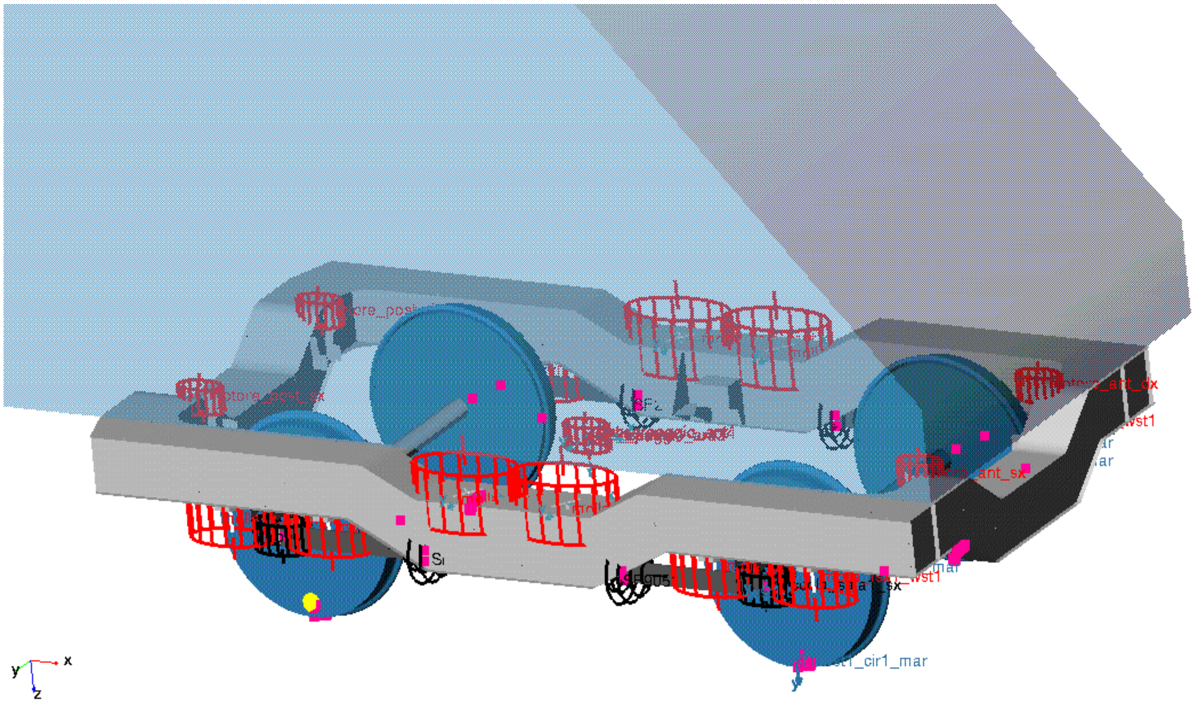


Figure 3 Front part of the model.

2.2 VALIDATION OF THE MODEL

The results of ADAMS' simulations were compared to those of the multibody code AD.Tre.S, developed at the Politecnico di Milano [3,14]. This code (AD.Tre.S) uses modal, finite element and multibody approach to describe the vehicle and it has been validated by experimental data [15].

ADAMS' results relative to the running on irregular track (using level IIa of the contact model [16]) showed a good agreement with the AD.Tre.S ones (Fig 4):

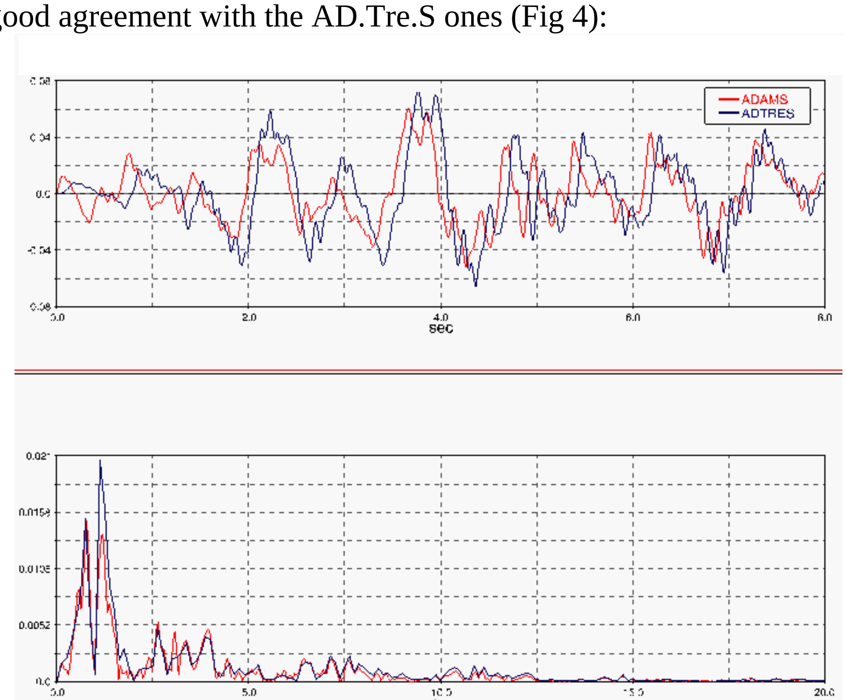


Figure 4 Comparison between ADAMS' and AD.tre.S.' results for vertical acceleration of the car body [m/s^2] (straight track with irregularity).

Simulations relative to the train running on a curve track (using level III of the contact model [16]) didn't show the same agreement and drove to consider only working situations not implying heavy non-linearity in the contact forces. That is the reason why the study presented in this paper was limited to the running on irregular track.

2.3 STRESS OUTPUT

The method was applied to the axle and to the bogie of the E412 locomotive with irregular straight track. An example of a time history of the stresses in a welding of the bogie and stresses in the whole bogie are presented in Figure 5: in these conditions the maximum value of dynamic stresses is about 30 % higher than static value.

Some problems have been encountered in the study of the bogie's stresses because of hardware resources and high simulation time. For this reasons, the entire methodology has been applied only for the wheelset, arriving to determine probability of failure of the axle relative to the running on irregular track.

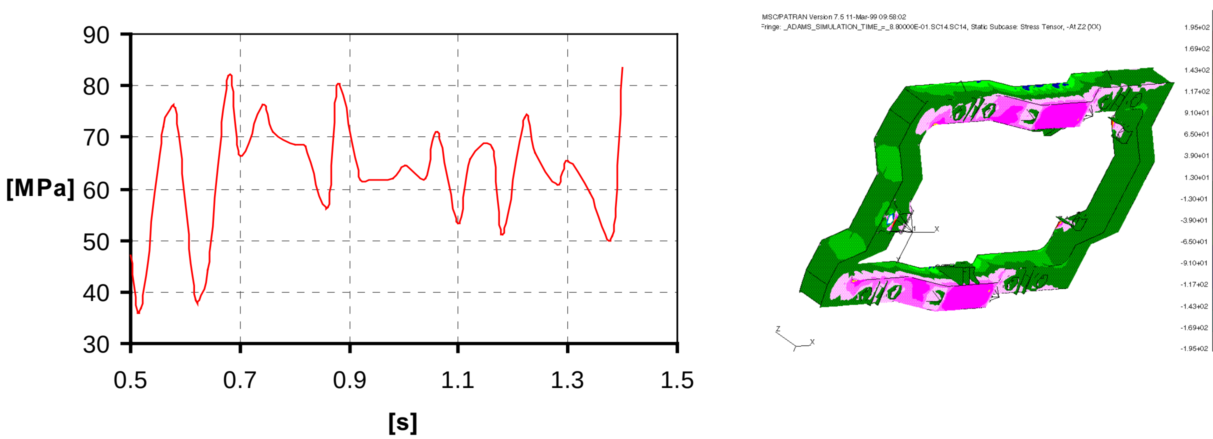


Figure 5 Stress time history in a welding of the bogie and stress distribution for a single step of simulation.

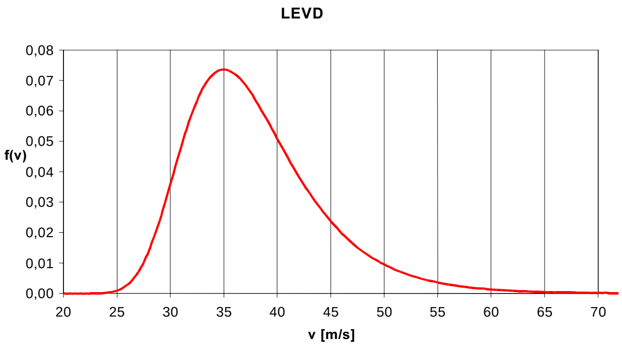
3. ESTIMATION OF STRESS SPECTRA FOR A VEHICLE RUNNING ON A STRAIGHT TRACK

3.1 SIMULATION

Fatigue life was calculated from the simulation of the vehicle on irregularity, using a statistical function describing the velocities. In the simulation has been assumed the *Largest Extreme Value Distribution (LEVD)* [4,5], characterised by two parameters: the scale parameter assumed λ=35 m/s and the shape parameter δ=5 m/s.

To each velocity is associated a probability of occurrence in straight run on irregularity and this probability is determined from the form of the statistical curve described before: assuming 3 Km of simulation, to each velocity corresponds a quote of 3 Km, related with its probability of occurrence.

The statistical curve of velocities has been sampled and only some points included in the range $\lambda-2\delta < v < \lambda+4\delta$ are considered: to each velocity may correspond a simulation. In Figure 6 is represented LEVD curve and simulation times for each velocity.



Velocity [m/s]	P[i-1<y<i] %	Distance [m]	Simulation time [s]
25	0.061798	1.8539313	0.074157252
30	6.537006	196.11017	6.537005686
35	30.18914	905.67422	25.87640617
40	32.43212	972.96356	24.32408898
45	18.12224	543.66717	12.08149273
50	7.800897	234.02692	4.680538464
55	3.041908	91.25724	1.659222554

Figure 6 LEVD distribution of velocities.

3.2 ESTIMATION OF THE SPECTRUM LOAD FOR 300.000 Km

The stresses acting on the wheelset may be determined using de Saint Venant's theory and in the Figure below is presented a typical time-depending trend of the stresses in a section of the wheelset comparing dynamic and static effects.

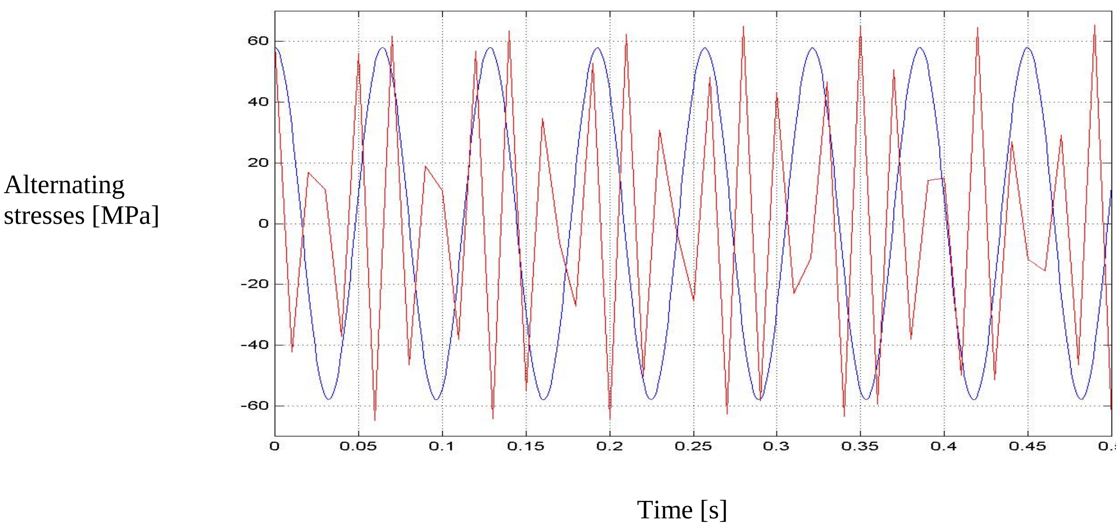


Figure 7 Time-depending trend of the stresses in a section of the wheelset : static BLUE, dynamic RED

The amplitude $\Delta\sigma_{alt}$ and the mean σ_{mean} of every hysteresis loop are calculated using rainflow's [6] cycle counting theory. From each simulation (for each velocity) may be determined a different load spectrum, and each one of this spectra gives its contribution to the total spectrum, that is the sum of the spectra calculated for each velocity.

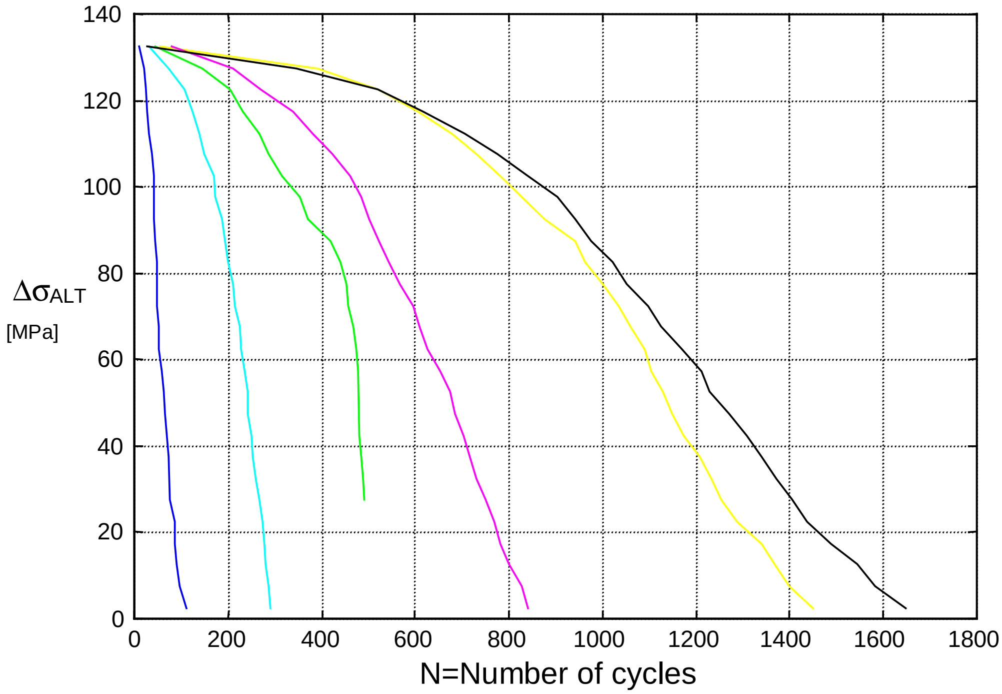


Figure 8 Load spectrum at the different speeds considered
V=30(GREEN);35(BLACK);40(YELLOW);45(MAGENTA);50(CYAN);55(BLUE)

The total spectrum load calculated is represented in Figure 9:

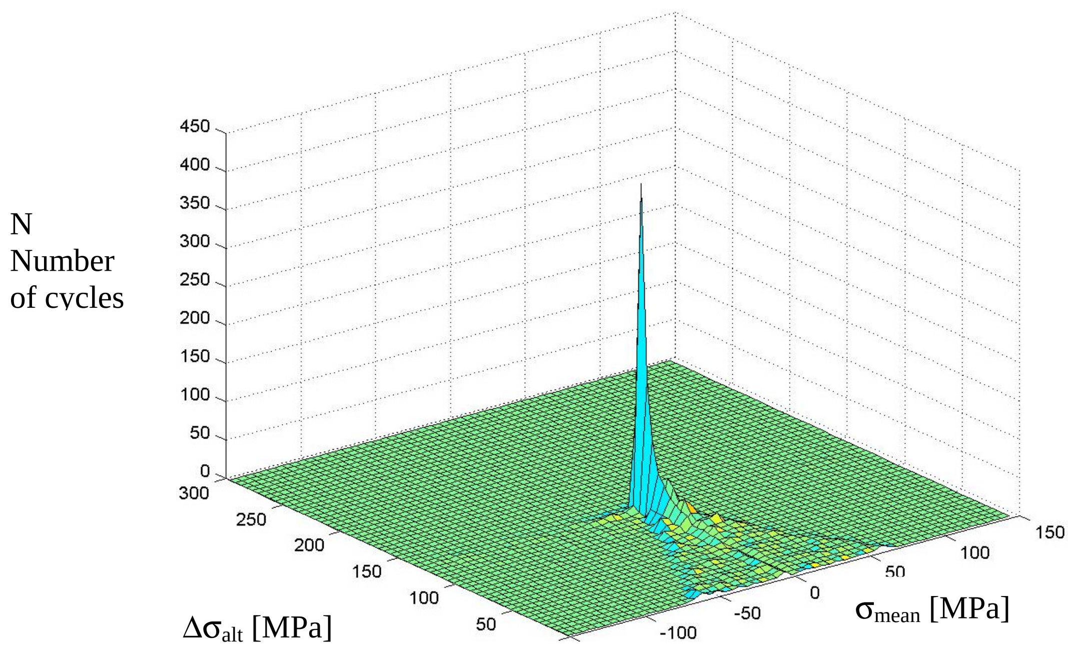


Figure 9 Representation of the total load spectrum.

From the form of this spectrum may be noticed two important informations: the highest alternate component of the stresses have a mean value near to zero, while lowest values of the alternate component of the stresses have a mean component far from zero value. Fatigue life calculations may be done assuming R = -1 (alternating fatigue) and so Murakami's [4,7,8] hypothesis and formulas may be used for calculating fatigue strength (Par 3.3).

For components of great size like bogies or wheelsets must be considered the presence of defects that shall be present in them. This defects should be measured and this measures should be studied statistically, so that can be found their statistical distributions. This is very important to use the formula purposed by some authors [4,7,8] for determining fatigue stress limit, but this way needs the knowledge of the distribution of measured defects.

In this work no measure on defects was available, so it was assumed to have noticeable defects [9,10] measured with non-destructive controls [11].

The statistical distribution of the spectrum load relative to 3 Km of running distance has been interpolated and it has been found to be the sum of two normal distributions. In order to obtain a spectrum load for a more significant running distance (300.000 Km), the spectrum relative to 3 Km has to be moved ahead of 10^5 number of cycles in a logarithmic scale and the remaining part before 10^5 cycles may be determined using the statistical distribution of the spectrum relative to 3 Km and the definition of mean time of occurrence.

An important validation the entire procedure has been done confronting the spectrum obtained from simulation and the spectrum obtained from measurements [2]; the results are quite satisfying being the spectra similar.

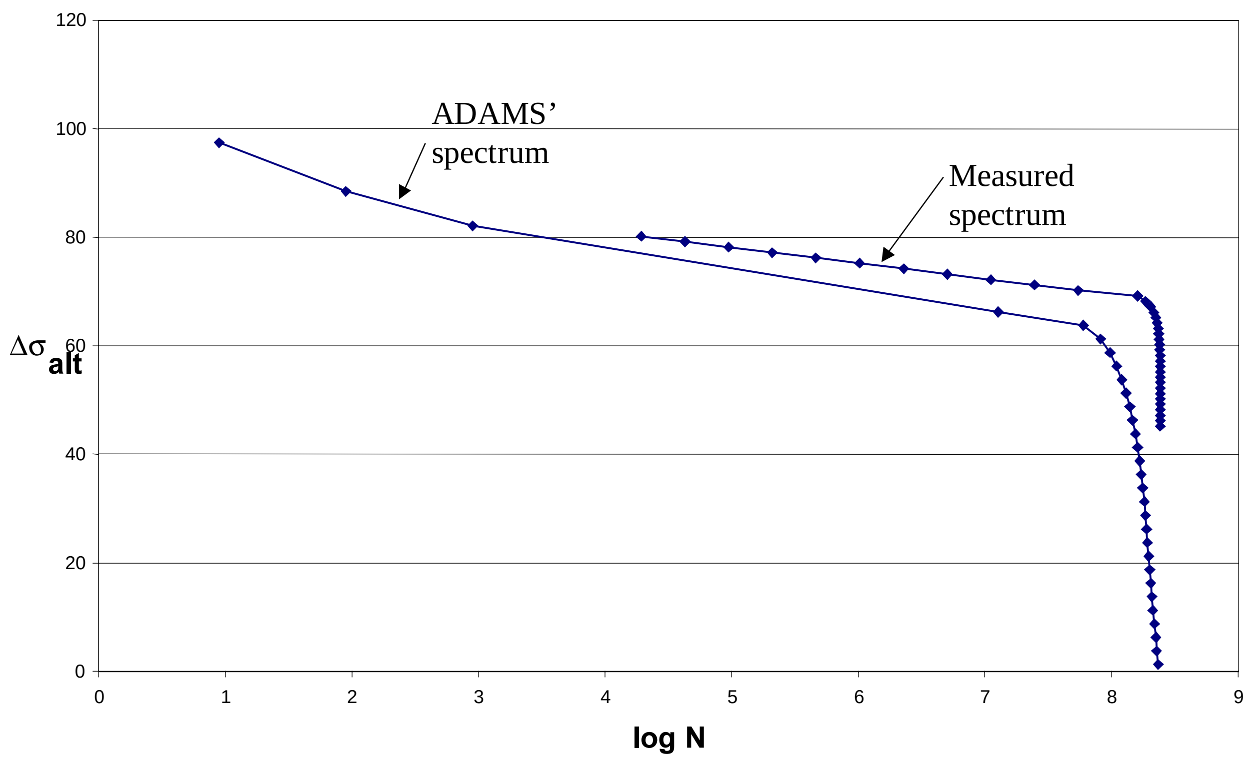


Figure 10 Total load spectrum versus N number of cycles: comparison between ADAMS' results and measurements.

3.3 FATIGUE LIFE INVESTIGATION

Murakami's formula [4] for determining fatigue strength is based on the assumption of the distribution of defects that may be present in the material, and links it also to material characteristics:

$$S_{LIM} = \frac{1.43\cdot(H_V + 120)}{\sqrt{area}^{\,1/6}} = S_{LIM}(\mu,\sigma) = S_{LIM}(\sqrt{area})$$

where S_{LIM} is the fatigue stress limit, H_V Vickers' hardness and $\sqrt{area}$ a characteristic dimension of defects, μ and σ are respectively the mean and the variance of the distribution of S_{LIM}.

The S-N curve has been determined using Haibach's [4] hypothesis, varying the fatigue stress limit S_{LIM}. The probability of failure has been calculated using damage hypothesis [12,13] for continuos (1) and discrete functions (2):

$$D = \sum_{i=1}^{N} \frac{n_i}{N} = 1 \qquad (1)$$

$$D = \int_0^\infty \frac{N \cdot p(s) \cdot ds}{C/S^m} = \frac{N}{C}\int_0^\infty S^{m\cdot}\, p(s)\cdot ds \qquad (2)$$

Fatigue life resistance has been determined multiplying the stresses of the load spectrum for a coefficient, called 'amplification coefficient of the stresses': varying this coefficient the probability of failure may be determined comparing the S-N curve with the spectra calculated (Figure 11).

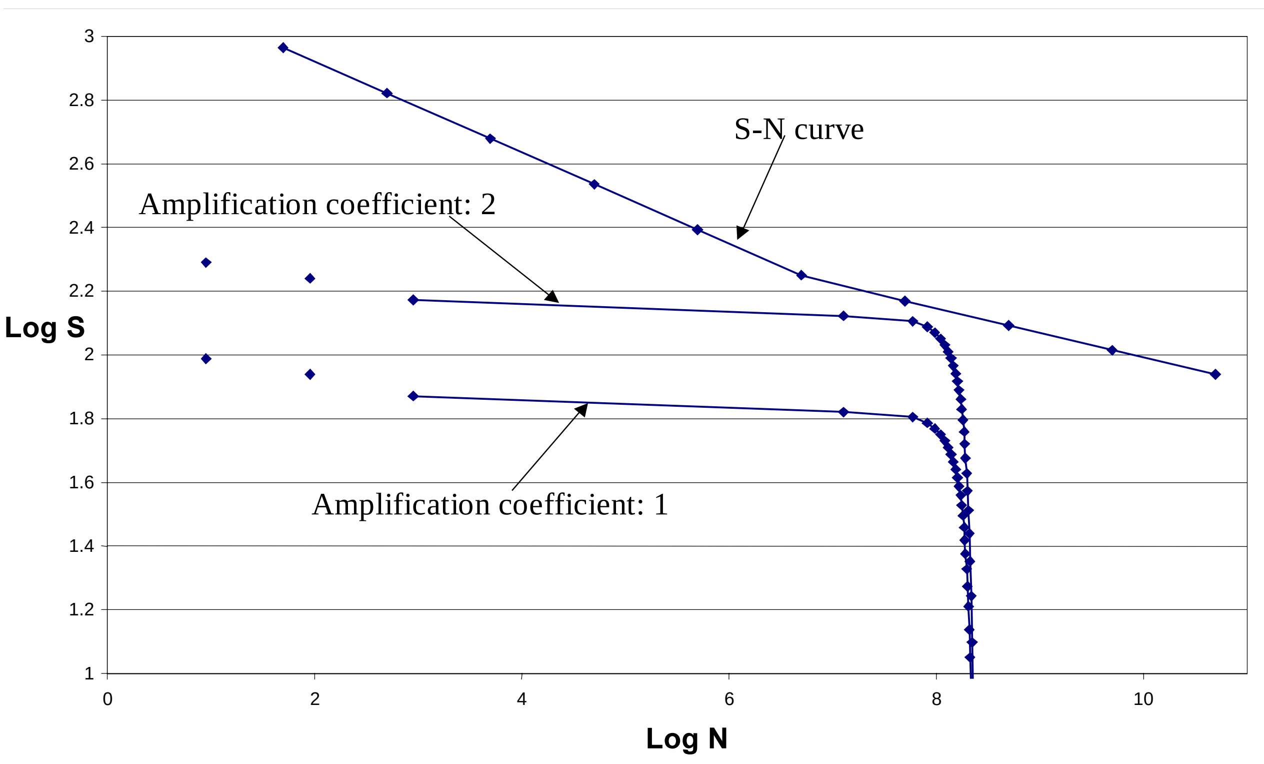


Figure 11 Determination of damage function confronting spectrum load and S-N curve

The results considering different amplification stresses are presented in the following table:

Safety factor $k = \dfrac{S_{LIM,\infty}}{S_{\max,N=1000}}$	*Amplification coefficient of stresses*	*Probability of failure % in straight track*
2,403	1	1.1e-31
1,612	1,5	0,00022
1,491	1,6	0,0065
1,438	1,7	0,094
1,335	1,8	0,75
1,193	2	0,805

Table I: Straight irregular track, distance 300.000 Km, probability of failure of the wheelset

CONCLUSIONS

A numerical model with 198 d. o. f. of the ADtranz E412 locomotive has been generated using the multibody program ADAMS/Rail 9.04, in order to investigate fatigue resistance in some point of the vehicle. The methodology purposed in this work for fatigue life investigation may be applied for any component of the vehicle (wheelsets, bogie, etc.).

Stresses and load spectrum may be calculated starting from a numerical model instead of making tests on the real vehicle: this is very important because allows to keep into accounting the dynamic stresses in the design phase, with economical benefits and advantages.

Load spectra for straight irregular track and traction has been calculated for 300.000 Km of distance and, comparing the spectra obtained with the S-N curves calculated assuming a population of defects, fatigue life resistance has been calculated with damage hypothesis.

The comparison with some measured spectra and the probability of failure calculated for the wheelset (compared with literature) have been quite satisfying, though probability of failure is very low. This is due to the fact that most significant conditions for fatigue evaluation of railway components are the passage over switches and cornering [2], that were not considered in this study.

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