FATIGUE LIFE SIMULATIONS APPLIED TO RAILWAY BOGIES

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ABSTRACT

The method described here is completely based on computer simulation. Finite element analysis is well established in the field of stress calculations. However, reliable information about loads acting on vehicle components such as bogie frames are only available for static situations up to now. Dynamic loads are often approximated by considering load factors, which are based on tests with realistic vehicles and which extend static load cases. Since the dynamical behaviour of railway vehicle systems changes with their evolution, approximations of dynamic loads by load factors might become obsolete and even prevent new trends as for instance lightweight constructions, see [1] and [2]. Multibody system analysis of the complete vehicle system allows the calculation of dynamic loads as input for finite element analysis so that stress and fatigue strength calculations can be significantly improved. Such a method makes it possible to consider different designs of the bogie frame and the complete vehicle system in an early stage of design and helps to obtain an optimum mechanical design.

Keywords: concurrent engineering, finite element analysis, multibody system analysis, fatigue life calculations

1. INTRODUCTION

The mechanical design process has to meet the specifications with respect to functionality, operating and manufacturing costs, energy consumption and environmental constraints, especially in the field of vehicle system dynamics. On the one hand lightweight constructions become more important as the energy consumption of (railway) vehicles increases with their weight. On the other hand the trend to lightweight constructions causes increasing structural vibrations and stresses, see [3]. Therefore, it becomes more and more important to consider interactions between different fields of engineering science such as computer aided design, multibody system analysis [4], finite element analysis and control system design. The application presented here, deals with the interactions of structural mechanics and vehicle system dynamics and their respective FEA and MBS software packages.

Dynamical loads acting on load bearing vehicle components are input for fatigue strength analysis. Usually they are approximated by dynamical load factors which extend static load cases, although multibody system analysis is capable of providing more reliable information about these loads.

Therefore, it is useful to consider elastic body deformation in multibody system models and to calculate dynamical loads and inertia forces by means of multibody system analysis. As there is no need to build and test realistic prototypes, different designs for load bearing vehicle components can be compared in an early stage of development. Coupled finite element and multibody system calculations allows to perform a sufficient number of design iterations, a prerequisite for optimal structural design, see [5].

The first problem with coupled finite element and multibody system calculations is the long-time operation of railway vehicles comprising a multitude of operating conditions such as crossing switches, curves or riding on straight track sections of different quality. All these situations and their frequency must be taken into account in order to obtain realistic results. Thus a representative section of the track is rather long and leads to large computing times, contradicting with the intention to perform design iterations. A second related problem is the computational effort of stress calculations by means of realistic finite element models. This paper presents a solution to both problems.

2. SIMULATION OF THE WHOLE VEHICLE OPERATION

In [6] it has been proposed to divide the whole operation of a vehicle into so-called operating conditions. Operating conditions comprise data about their frequency, the excitation by the track and the vehicle speed. Based on this information multibody system analysis is once and separately performed for each operating condition. For example, time integration of the nonlinear multibody system model yields forces acting on the bogie frame when the freight locomotive passes a switch, whereas linear multibody system analysis yields a covariance matrix of forces due to excitations by track irregularities. Information about the number of switches and the length of the section of the straight track will be considered later during fatigue life calculations, see Fig. 1.

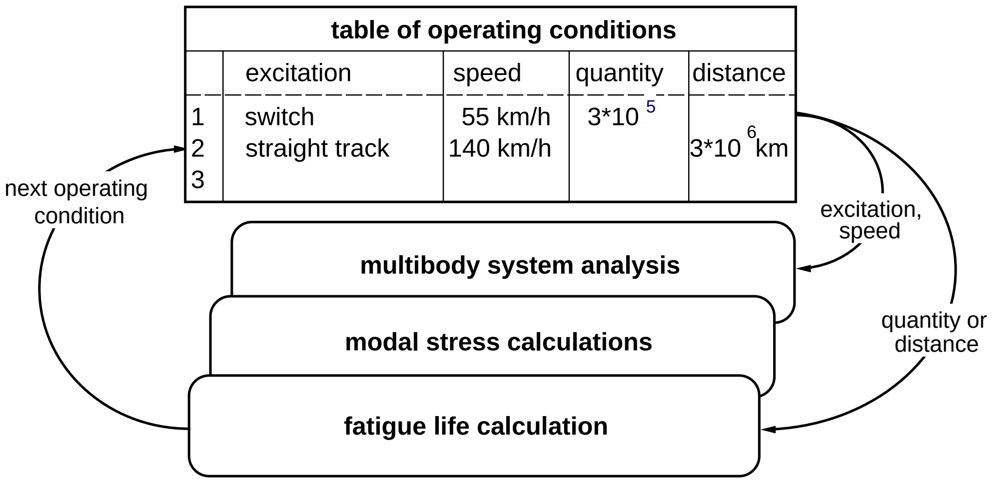

Fig. 1 Breaking down the vehicle operation into operating conditions.

3. FAST STRESS CALCULATIONS

In most cases elastic body representation in multibody systems is realised by Ritz-approaches [11]

$$\boldsymbol{u}(\boldsymbol{c},t) = \sum_{j=1}^{n} \boldsymbol{u}_j(\boldsymbol{c})\, q_j(t) \tag{1}$$

which are linear combinations of mode shapes $\boldsymbol{u}_j(\boldsymbol{c})$ weighted by their respective time dependent modal coordinates $q_j(t)$. Stress calculations based on Ritz-approaches - in short terms *modal stress calculations*

$$\boldsymbol{\sigma}(\boldsymbol{c},t) = \sum_{j=1}^{n} \boldsymbol{\sigma}_j(\boldsymbol{c})\, q_j(t) \tag{2}$$

depend on a number of modal coordinates, which is small compared to the number of degrees of freedom of finite element models. Stresses $\boldsymbol{\sigma}_j(\boldsymbol{c})$ can be calculated by the linear strain tensor $\boldsymbol{D}_{\varepsilon u}$ and by Hooke´s law $\boldsymbol{C}$ from the modes $\boldsymbol{u}_j(\boldsymbol{c})$

$$\boldsymbol{\sigma}_j(\boldsymbol{c}) = \boldsymbol{C}\, \boldsymbol{D}_{\varepsilon u}\, \boldsymbol{u}_j(\boldsymbol{c}). \tag{3}$$

However, the modal approach yields good results for deformations $\boldsymbol{u}$, whereas the rate of convergence is poor for stresses, see [8] and [6]. Therefore, a sufficient Ritz-approximation of stresses demands for the consideration of a large number of modes and modal coordinates, respectively. Since modal coordinates are state variables of elastic multibody systems, former modal stress calculations increased the computational effort of multibody system analysis, see [8] and [6]. Furthermore, the Ritz-approach comprises mostly eigenmodes, which were selected by estimations on elastic body deformation – in other words by trial and error.

The rate of convergence can be significantly improved by means of *static modes*. In order to obtain static modes loads $\boldsymbol{p}(\boldsymbol{c},t)$ acting on the bogie frame are represented by the Ritz-approach

$$\boldsymbol{p}(\boldsymbol{c},t) = \sum_{k=1}^{n} \boldsymbol{p}_k(\boldsymbol{c})\, q_k(t), \tag{4}$$

which comprises loads $\boldsymbol{p}_k(\boldsymbol{c})$ representing forces acting in suspension elements as well as inertia loads. The basic idea is to compute mode shapes or static modes $\boldsymbol{u}_k^p(\boldsymbol{c})$ corresponding to loads $\boldsymbol{p}_k(\boldsymbol{c})$ and to consider these static modes in combination with eigenmodes in the Ritz-approaches (1) and (2). Finite element models, represented by the mass matrix $\boldsymbol{M}$ and the stiffness matrix $\boldsymbol{K}$, yield static modes[1] for constant frequencies Ω_0

$$(\boldsymbol{K} \;-\; \Omega_0 \boldsymbol{M})\, \boldsymbol{u}_k^p(\boldsymbol{c}) \;=\; \boldsymbol{p}_k(\boldsymbol{c}). \tag{5}$$

A sufficient rate of convergence within a frequency band $\Delta\Omega$ is guaranteed by considering eigenmodes. These eigenmodes can be selected in a deterministic manner by means of the criterion

$$q_j \;=\; \frac{(2\Delta\Omega\Omega_0 \;-\; \Delta\Omega)}{(\omega_j \;-\; (\Omega_0 \;+\; \Delta\Omega)^2)} \;\; \frac{\boldsymbol{u}_j^{h\;T}\; \boldsymbol{M}\, \boldsymbol{u}_k^p}{\boldsymbol{u}_j^{h\;T}\; \boldsymbol{M}\, \boldsymbol{u}_j^h}, \tag{6}$$

where eigenmodes $\boldsymbol{u}_j^h(\boldsymbol{c})$ corresponding to large q_j values have to be considered in the Ritz-approximations (1) and (2). Thus, all modes of the Ritz-approach can be selected based on information about interconnections of the elastic body (bogie frame) with the remaining vehicle system and the requested frequency range of the actual application. Since this information is given, the Ritz-approach is established in a deterministic manner but not by trial and error.

It is possible to perform modal stress calculations for all locations of a structure, but in most cases the resulting stress distributions will show locations which are heavily stressed under many different loading conditions, whereas other regions are almost unstressed. Therefore it is useful to perform modal stress calculations only for those critical locations. A procedure for their localisation will be described below. This leads to the stress load matrix concept. For critically stressed locations $\boldsymbol{c}_{cr}$ the stress load matrix

$$\boldsymbol{B}(\boldsymbol{c}_{cr}) = [\boldsymbol{\sigma}_1(\boldsymbol{c}_{cr}), \cdots, \boldsymbol{\sigma}_n(\boldsymbol{c}_{cr})] \begin{Bmatrix} q_1(t) \\ \vdots \\ q_n(t) \end{Bmatrix} \tag{7}$$

contains column by column stresses which correspond to the modes of the Ritz-approach (1). Since the number of rows and columns of the stress load matrix is small compared to the

[1] The denotation static mode is based on reduction techniques such as Guyan reduction or the Craigh Bampton method, which use static modes (constraint modes) for a frequency of zero, see[1].

number of degrees of freedom of the finite element model, the stress load matrix enables stress calculations in low computing times. The stress load matrix concept can not only be applied to output quantities $q(t_i)$ of nonlinear multibody system analysis

$$\boldsymbol{\sigma}(\boldsymbol{c}, t_i) = \boldsymbol{B}(\boldsymbol{c}_{cr})\, \boldsymbol{q}(t_i) \tag{8}$$

it is also applicable to output quantities of linear system analysis such as covariance matrices $\boldsymbol{P}(\boldsymbol{q})$

$$\boldsymbol{P}(\boldsymbol{\sigma}) = \boldsymbol{B}(\boldsymbol{c})^T\, \boldsymbol{P}(\boldsymbol{q})\, \boldsymbol{B}(\boldsymbol{c})\,. \tag{9}$$

In the first case one obtains stresses $\boldsymbol{\sigma}(\boldsymbol{c}, t_i)$ at times t_i, while covariance analysis yields variances and covariances of stresses $\boldsymbol{P}(\boldsymbol{\sigma})$. In this way the stress load matrix concept enables further reductions of computing times, because it allows the combination of linear and nonlinear multibody system calculations.

4. FATIGUE LIFE CALCULATIONS

The next task is to evaluate stresses and compare them with stress limits as known from fatigue strength tests. Since fatigue strength tests are based on processes with a harmonic dependence of time but vehicle components are irregularly loaded, fatigue life calculations transform irregular processes as given by equations (8) and (9) into an *equivalent harmonic process*. This reduction will be performed in three steps, which are based on the assumption of the equivalence of damage.

In a first step, amplitudes and mean values of damaging stress cycles are counted and stored in a Markov matrix. Since comparisons between calculation and experiment shows a good correspondence, the rainflow method is a well established method for counting stress cycles in the time domain , see [9]. The number of stress cycles stored in Markov matrices can be multiplied with the number excitations such as crossed switches, compare with Fig. 1.

The evaluation of the results of linear system analysis as for instance the covariance matrix of stresses (9) will be detailed now. A procedure as proposed in [9] will be applied. Consider a probability distribution function describing mean stress components by a Rayleigh distribution whereas the distribution of mean stresses is represented by a Gaussian distribution. Combining both by multiplication yields a distribution function, whose integration yields the number of stress cycles per second, see Fig.2

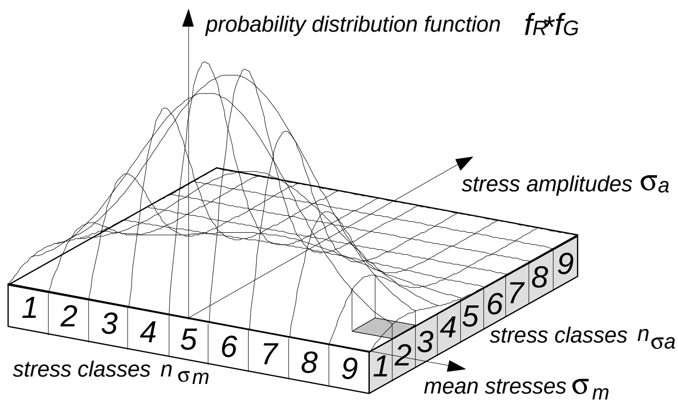

Fig.2: Probability distribution function

Subdividing into stress classes and separately performing integrations for the stress classes one obtains a matrix which is equivalent to a Markov matrix, so that the distinction between different kinds of MBS-output is unnecessary for the following evaluations.

The elimination of mean stress components is performed by extended amplitude transformation, which takes the damaging effect of mean stress components of stress cycles into account. It is known for sinusoidal processes, that stress amplitudes decrease linearly for increasing mean stress components, see [10]. Assuming this linear relation also for stochastic processes one obtains the extended amplitude transformation

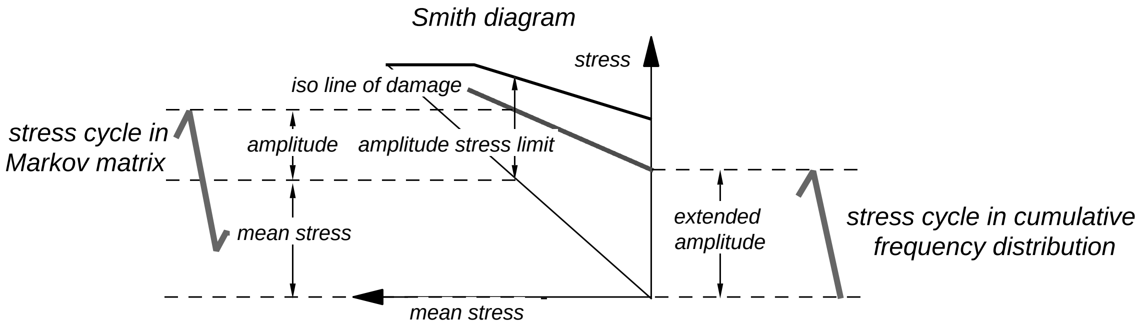

Fig.3: Extended amplitude transformation results in cumulative frequency distributions.

which allows the elimination of the mean stress component. The sensitivity of stress limits to mean stresses for stochastic processes was investigated in [10]. Extended amplitude transformation leads to usual one dimensional cumulative frequency distributions, for which damage values can be calculated as described in [9].

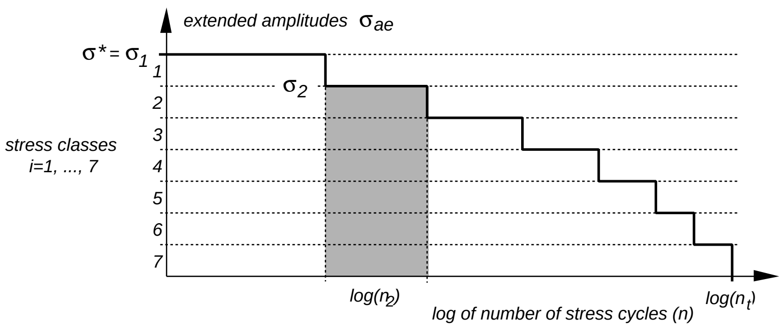

Fig.4: Cumulative frequency distribution

Cumulative frequency distributions are described by the maximum value of stresses σ^* and number of stress cycles n_i corresponding to the stress classes *i*. The total number of stress cycles is given by n_t. As described in [9], equivalent harmonic stresses (11) can be calculated by means of the shape factor v_k of the cumulative frequency distribution

$$v_k = \sqrt[6]{\frac{n_i}{n_t}\left(\frac{\sigma_i}{\sigma^*}\right)^6}, \qquad (10)$$

the quotient of the number of stress cycles N_D corresponding to the endurance limit and total number of stress cycles n_t

$$\bar{\sigma} = \frac{1}{v_k^{-0.787}\left(N_D/n_t\right)^{0.154}}\sigma^*. \qquad (11)$$

Damage values (12) for operating conditions can be calculated with equivalent stresses $\bar{\sigma}$, static stresses σ_s and fatigue stress limits σ_l

$$D = \frac{\bar{\sigma} + \sigma_s}{\sigma_l} \qquad (12)$$

Damage values (12) can be superimposed to a total damage value by means of Cortan´s and Dolan´s cumulative damage hypothesis, see [9].

5. COUPLED FINITE-ELEMENT-AND MULTIBODY SYSTEM ANALYSIS APPLIED TO THE BOGIE FRAME OF A FREIGHT LOCOMOTIVE

After time integration of the multibody vehicle system has been performed, short sections of the time series of MBS-output quantities can be transferred back to the FEA-code ANSYS or NASTRAN by the MBS-postprocessor LOADS . Quasi-static finite element analysis and the graphical representation of stress distributions shows maximally stressed locations, see Fig.5 b).

a) b)

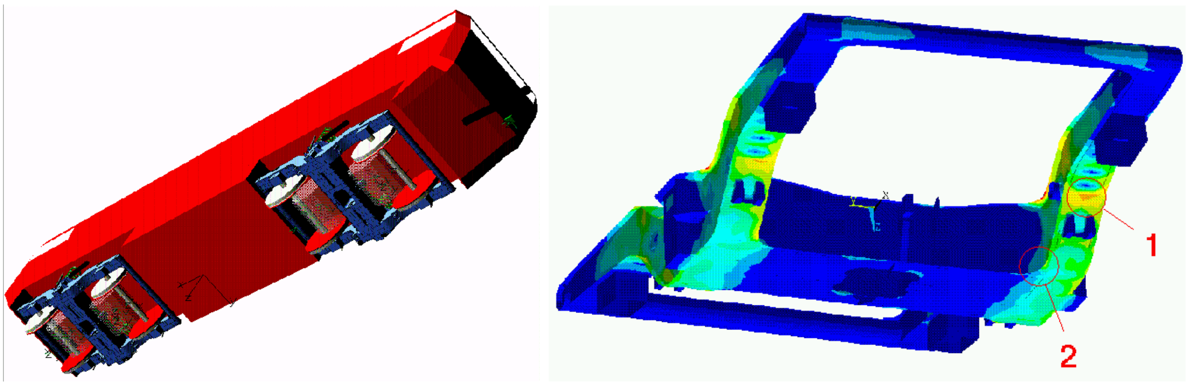

Fig.5 : a) MBS model of the investigated freight locomotive, b) FE model of the corresponding bogie frame; distribution of Mises stresses, when the leading wheelset gets into contact with the common crossing of the switch.

As an example for a nonlinear operating condition the freight locomotive taking the divergent route through a switch with constant speed 55.0 km/h is considered. Fig. 6 a) shows a top view of the switch with its variable rail profiles of the common crossing and the check rail. At time t = 1.63 sec the foremost wheelset passes the switch blade with double contact on the left wheel. The second set of oscillations arises when the wheelset goes through the common crossing on the left wheel and gets into contact between the back of right wheel and the check rail at t=3.67 sec. For those two times all forces and the absolute motion were transferred back to ANSYS using the LOADS program. Quasi-static finite element analysis for t = 3.67 sec results in the stress distribution as shown in Fig. 5 b), which shows some critical stressed locations at the bent part of the side frames and the transitions between side frames and the transom. Two of those locations were selected for fatigue life calculations and Fig. 6 b) shows corresponding time histories of normal stresses in x-direction obtained by a stress load matrix.

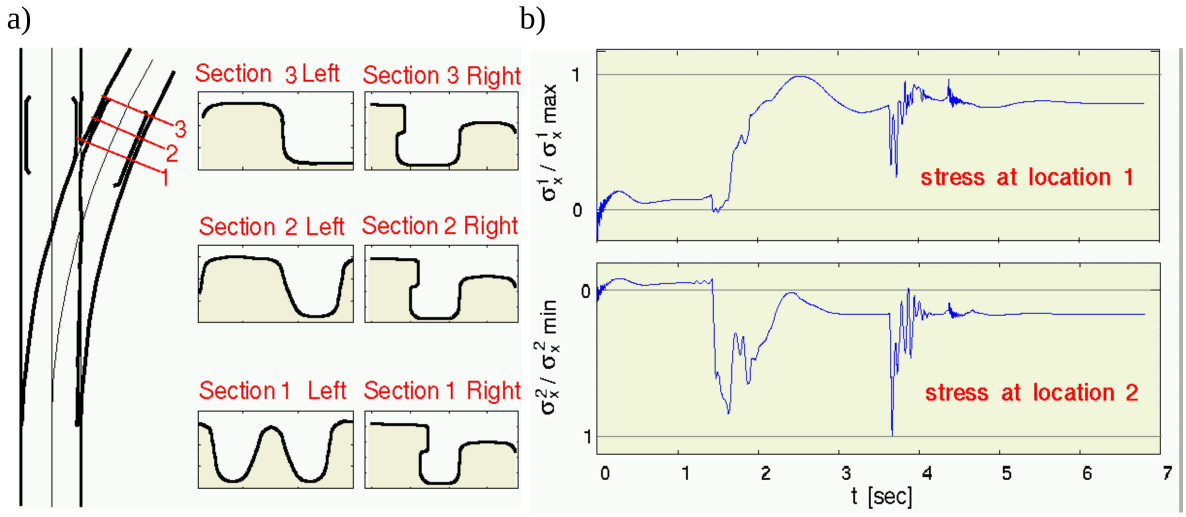

Fig.6: a) Top view of the switch with its variable geometry of the common crossing and the check rail, see [11]; b) Normalised longitudinal normal stresses for locations 1 and 2, see Fig 5 b).

In contrast, riding on straight track was investigated by covariance analysis. The vehicle runs with its reference speed of 140 km/h on the straight track. The track excitation is defined by standard power spectral density functions for the vertical the lateral and the cross level track irregularities. A covariance matrix for vertical forces acting in the primary and secondary suspension was transformed by the stress load matrix into stresses and stress evaluations finally gave the damage values as listed in table 1. The results correspond to the table of operating conditions as shown in Fig. 1.

critical location	Normalised Damage Values for passing switches	riding on straight track
1	0.90	1.00
2	0.48	0.61

Table 1: Damage values

Fig. 6 b) shows a quite small number of stress cycles and in combination with the assumption that a switch occurs every 10 km small damage values are plausible, although the maximum dynamic load in vertical direction of 17.5 KN is large compared to the maximum dynamical RMS value of the vertical force of 3 KN which acts in the primary suspension when the locomotive runs on the straight track. The results are expected to change for an improved track quality.

6. SUMMARY AND CONCLUSIONS

The interactions of structural and vehicle system dynamics with respect to fatigue life calculations on condition of short computation times were discussed. The new method is applicable to realistic finite element and multibody system models not only in the field of railway engineering. It assesses also the development of new airframe and road vehicle structures. A combination of linear and nonlinear multibody system analysis as well as fast modal stress calculations enables shortest computing times and the possibility to consider a multitude of operating conditions and their frequency. As the vehicle operation and the dynamical behaviour of the vehicle are taken into account, the reliability of fatigue life calculation of vehicle components is significantly improved by the strategy proposed here. The new method enables to compare different designs of the bogie frame and helps to obtain an optimum of structural design, especially when new technologies are requested.

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