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RESPONSE SPECTRUM ANALYSIS OF LARGE VEHICLE SYSTEMS

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Abstract

The dimensioning of vehicle body structures for service fatigue life is a highly complicated task in its every stage. Namely the appropriate structural modelling for dynamic analysis and the elaboration of realistic loading and design conditions (loading and design spectra) for the total duration of their life. In this paper the dynamic analysis of a bus is presented as a feasibility study using finite element model of large number of degrees of freedom.

Keywords: large vehicle models, modelling by finite elements, dynamic analysis, simulation of road profiles, evaluation of response spectra.

1. Introduction

The failure of mechanical structures due to material fatigue is usually originated from local yields (dislocations) in the material. Prediction of fatigue life of one or more elements of a large mechanical system necessitates the precise knowledge of the position and process (in time) of local stress concentrations producing errors or deterioration of it. This fact demands the application of well-detailed structural, usually finite element models of large number of degrees of freedom. A number of Hungarian researchers have successfully studied the theoretical, computational and measuring aspects of this problem ([1], [2], [3]), etc.). However, the actual calculations have been carried out on smaller mechanical models since earlier there were no satisfactory computational possibilities. Nowadays some developments can be observed in this area namely some of the professional finite element programs are currently available (NASTRAN, COSMOS/M, ANSYS, etc.). When the number of degrees of freedom of a finite element model is about some thousands the most useful way is the application one of these finite element programs.

Each phase of strength calculation of vehicle body structures for service fatigue life is a very complex and complicated task. First phase is the determination of representative sets of loads that are valid for the total duration of life of vehicles. These loads, for example, originate from the roughness of different kinds of roads, manoeuvres like steering, acceleration or braking and from the excitation of the engine and power transmission. Moreover the payload is usually changed during the service life of a vehicle. Second phase is the elaboration of an appropriate vehicle structural model. Actually a vehicle body can be described as damped linear elastic system while the behaviour of suspensions and tyres are non-linear (damping and stiffness characteristics). Having determined the required vehicle responses the last phase is the fatigue life calculation itself.

For the reliable strength calculation for service fatigue life of vehicles experimental data are indispensable. Considering the input loads it is necessary to know the (measured) excitations of different roads and their expected rate of occurrence during the vehicle life of duration. Besides the road roughness measurements there are publications in the modelling of road profiles and surfaces ([4], [5]) since it is not so easy to measure parallel tracks below left and right wheels simultaneously. Moreover the designers are much more interested in the expected behaviour of vehicles over a large number of roads of the same class than in their detailed behaviour on a particular road. In the second phase especially the determination of the stiffness and damping characteristics of tyres as well as the damping of vehicle bodies requires measured data. At last, in the third phase, the elaboration of design fatigue curves requires experiments [6].

In this paper dynamic analysis of a bus structure is carried out by finite element method using the COSMOS/M finite element program. The number of degrees of freedom of the applied finite element model is 1852. Excitations are derived from two-dimensional power spectral density function which describes the roughness of road surface in vertical direction.

2. Simulation of Road Excitations

Measuring parallel road profiles simultaneously is always a very complicated and difficult operation. Final (road profile) data from measurements are usually carried out indirectly after filtering, signal analysis and unavoidable data transformations. Therefore there are numbers of attempts for the spectral representation of road surfaces ([4], [5]). In the road surface simulation we follow the method contained by paper [5] in which it is proved that from

$$G(n_x, n_y) = \frac{0.5G_0}{\sqrt{n_x^2 + n_y^2}}, \tag{1}$$

the two dimensional power spectral density (psd.) function, the next one dimensional psd. function can be derived for random description of road profiles

$$G_1(n) = \frac{G_0}{n^2}. \tag{2}$$

In the previous equations G_0 is constant moreover n_x, n_y and n are spatial wave numbers.

In paper [4] for the random representation of road profiles the relationship

$$G_1(n) = \frac{G_0}{n^{2.5}}, \tag{3}$$

is suggested where $0.01 \leq n \leq 10$ cycles/m and values for G_0 are as follows,

motor way : $G_0 = 3 + 50 \times 10^{-8}$,
major road : $G_0 = 3 + 800 \times 10^{-8}$,
minor road : $G_0 = 50 + 3000 \times 10^{-8}$,

where $G_1(n)$ is the spectral density of road roughness in m^3/cycle and the n the unit of n is cycle/m. The unit of G_0 is compatible with other quantities in *Eq.* (3). It can be proved by direct calculation that the results in paper [5] are applicable for the *Eq.* (3). Having applied it we get the two dimensional psd. function

$$G(n_x, n_y) = \frac{G_0}{1.748\sqrt{n_x^2 + n_y^2}^{3.5}} \tag{4}$$

that will be used for the isotropic description of road surface roughness.

When a professional finite element program is applied the user is constrained by its possibilities. In case of the most finite element programs similarly to COSMOS/M the response spectrum analysis can only be performed for diagonal input spectrum matrix that is the cross spectra are assumed to be zero. Therefore it may not be applied directly for road surface excitations since cross spectra among left and right wheels are not negligible. This difficulty is overcome when the psd. functions of tracks below the wheels are represented by their realizations along the road in the function of driving distance. Using SHINOZUKA's method [7] the road profile realizations can be simulated from the psd. function given by *Eq.* (4) besides the value of $G_0 = 50 \times 10^{-8}$ which corresponds to major roads of better quality (*Fig. 1*).

3. Model Elaboration

The discussed finite element model shown in *Fig. 2* is elaborated on the basis of an actual bus. Frame structures of bus bodies usually have linear elastic properties, however, the behaviour of suspension systems and tyres is non-linear. In some cases these nonlinearities may not be neglected, for example in the case of stability problems or studying the effect of extreme road irregularities, etc. In other cases the characteristics of suspensions and tyres can be approximated by linear ones with acceptable errors when road vehicles travel on country roads of good or average quality with constant speed. Linearization of these characteristics is based on their nominal operating data released by manufacturers.

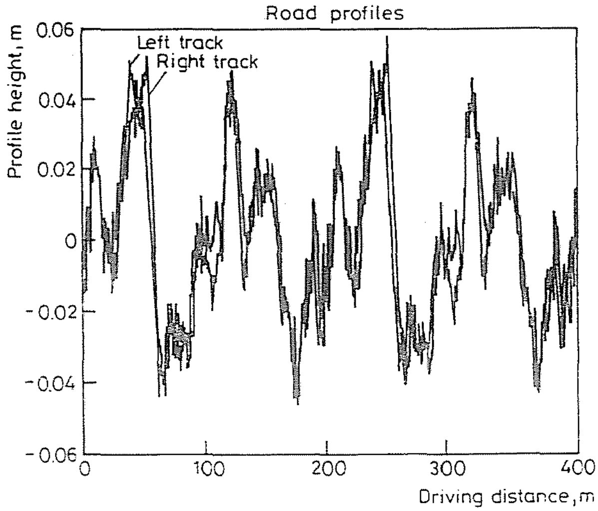

Fig. 1. Simulated road realizations below left and right wheels

The continuously distributed mass of the studied bus is divided into nodes in a manner that its mass matrix is a lumped one. As it was mentioned above the stiffness and damping of suspensions and tyres is approximated by linear characteristics while structural damping of bus body as Raleigh's damping is taken into consideration assumed to be proportional to the stiffness matrix. Kinematic excitations of road roughness in the function of time are derived from the simulated road profile realizations assuming constant travelling speed of 20 m/s. Time delay between front and rear wheels is considered.

Number of degrees of freedom of the studied finite element model is 1852 and the number of nodes and mass points is 325 and 207, respectively. The number of beam elements is 533 and 128 shell elements are built in the body of the bus model.

4. Dynamic Analysis and Results

In service, bus bodies among others are subjected to the vertical excitation of road surface roughness describing as a stationary random process. Actually the applied time history functions for excitations are derived from the road profile realisations of this random process. In compliance with it the response stress-time history functions can also be considered as realisations of a stationary random process. Being in the possession of fatigue design curves these stress time history functions can be used for the estimation of the average fatigue life [8].

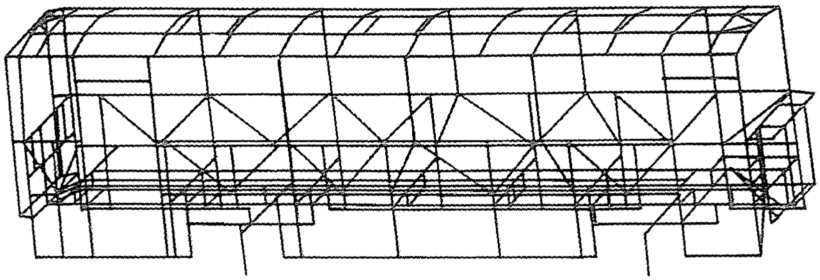

Fig. 2. Wire network sketch of the studied finite element model

Calculations of stress time history functions are carried out by the modal time history analysis module of COSMOS/M in two phases. In the first phase the lower 45 undamped natural frequencies and mode shapes are calculated up to 20 Hz. This upper limit of natural frequencies is enough to a correct dynamic stress analysis. Then, in the second phase, using these natural frequencies and mode shapes the stress-time functions are calculated in the required equidistant time points. The effect of the concentrated viscous dampers, built in the finite element model, is calculated in each time step by an iterative process. Similarly the material damping of the bus structure is taken into consideration during the second phase of the dynamic analysis. In *Figs. 3* and *4* the resultant of axial stresses (from bending moments and axial forces) and the shear stress (from torsion) can be seen respectively, arising at one end of a beam element located in the left side longitudinal web of the chassis of the bus. These stress-time history functions are made only for illustrations. In case of actual calculations the lengths of the considered time intervals can be increased to the required lengths.

If statistical nature of response stress records is necessary there is possibility to generate their power spectral density functions using Fourier transformations. For example in *Fig. 5* the power spectral density function of the axial stress record is demonstrated which is shown in *Fig. 3.* On the basis of power spectral density functions the standard deviations of response stresses can also be calculated. In case of the presented axial stress the magnitude of its standard deviation is 8.227×10^6 [N/m^2].

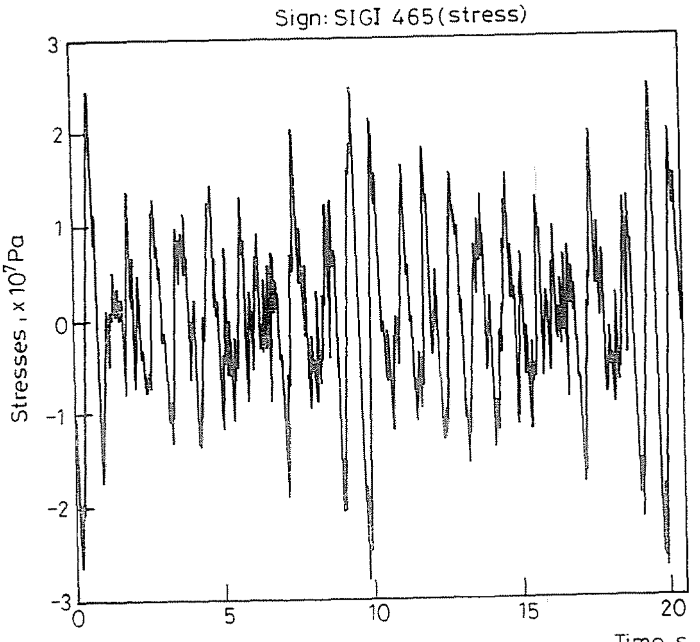

Fig. 3. Resultant axial stress

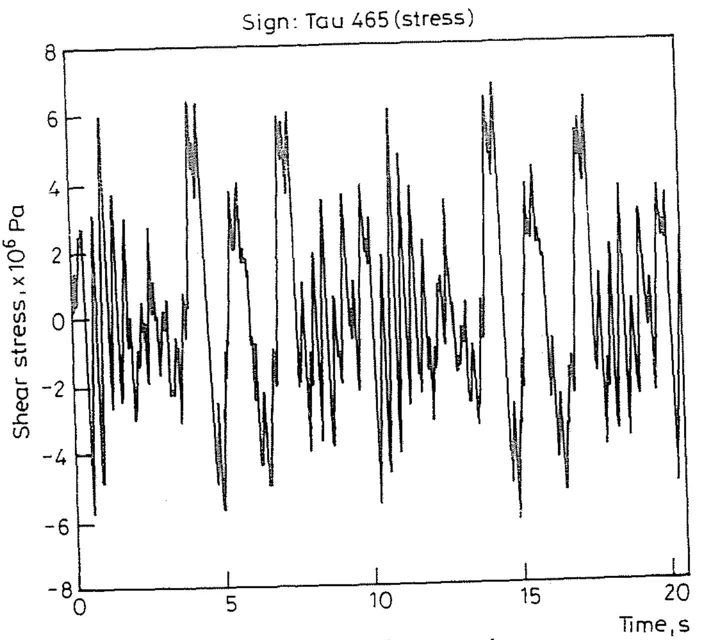

Fig. 4. Shear stress from torsion

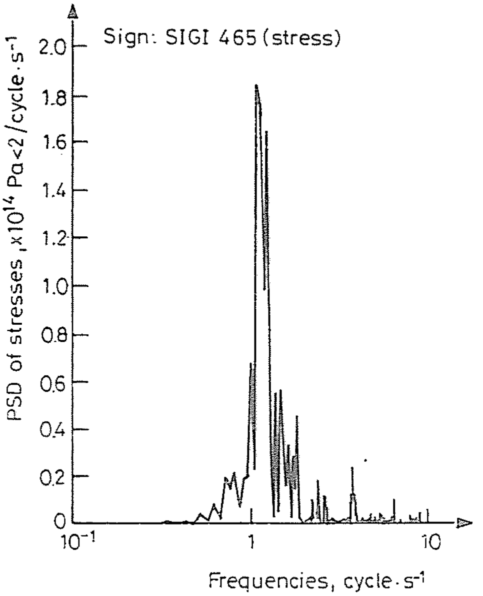

Fig. 5.

5. Conclusions and Future Tasks

The analysis presented in this paper has shown the potential applicability of one of the professional finite element programs for the calculation of fatigue life of large vehicle structures considering vertical excitations of road surface roughness.

In future there are a lot of problems to solve, for example:

- Determination of representative sets of roads that characterize the realistic road excitations for the total duration of life of vehicles. (By measurements and on the basis of literature data.)
- Determination by measurements of material damping in bus body structures.
- Theoretical and numerical study of the accuracy of linear approximation of the non-linear suspension and tyre characteristics, etc.
- Detailed modelling of the reaction of the passengers, etc.

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