SIMULATION OF DYNAMIC LOAD TRANSFER TOTHE STRUCTURAL NODES OF VEHICLE

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Abstract *In this paper the methodology of the development of a computationally efficient full vehicle virtual model is shown. The measured properties form the basis for the alignment of the properties of the virtual model with a real vehicle. The hysteresis of flexible elements, suspension arms and body work was taken into account. The correctness of the theoretic assumptions and modeling approaches was confirmed by the acceptable consistency of the results obtained by calculation, simulation and measurement. Such a model can be used for the rapid and reliable development of a vehicle in terms of handling, driving safety, ride comfort, noise emissions, vibration, the harshness of the operational functions, and the determination of dynamic load.*

1 INTRODUCTION

For the rapid development of a new type of vehicle with the required properties, it is necessary to have an effective virtual computing model [2]. The goal of this research is to elaborate a methodology to determine the degree of influence of individual elements, such as flexible mounts, shock absorbers, bumpers, suspension arms, the subframe and the white body, on the resulting dynamic performance of the vehicle [1]. A virtual model of a full vehicle created in accordance with the proposed methodology should have properties that correspond with the results of the measurements. Such a model then can be used to study the effect of rapid changes in the parameters on the resulting properties of the vehicle [4].

2 REALIZATIONS OF MEASUREMENTS

The static and dynamic characteristic curves of vehicle components (flexible storage, shock absorbers, bumpers, main spring, etc.) obtained from the measurements are shown in Fig. 1. To carry out these measurements it was necessary, in collaboration with the Slovak Academy of Sciences, to combine a measuring device with an electro-hydraulic device EDYZ4. The measuring chain consisted of a potentiometric position sensor, a force sensor, an output card for generating the excitation signal, an input card for collecting data from the encoder, an input card for data collection from the power sensor, control electronics and a computer. To conform to the necessary excitation frequency, it was necessary to correct the deviations using PID feedback control. Signal recording was done using LabView SignalExpress and its filtration was done using the program Matlab.

The measurement of the dynamic properties of the full vehicle was carried out in cooperation with Škoda Auto Mladá Boleslav on the four-post road simulator.

The locations of individual sensors are marked in Fig.2 by the following numbers: 1- sprung masses, 2-center of gravity, 3-engine, 4-gear, 5-measuring wheels and unsprung masses, 6-main springs, and 7-excitation cylinders.

Figure 1: Measurement of the static and dynamic characteristics of stiffness of: a) damper, b) flexible mount of damper and c) bump stop.

Figure2: The four-post road simulator

Figure3: The dark curve represents the left side of the front axle; the pale curve is for the rear axle. Upper figure: the course of the Power Spectral Density PSD of sprung masses; lower figure: the course of the PSD of the unsprung masses of a vehicle.

3 PROCESSING OF RESULTS FROM MEASUREMENTS

The eigenfrequencies of the individual parts of a vehicle, which are necessary for tuning the vehicle, can be recognized from the course of the PSD of the measured signals (see Fig.3) after their transfer from the time to the frequency domain. The eigenfrequencies $\omega_{\text{NH Par}}$ of the unsprung masses during vertical parallel travel can be obtained from the formula

$$\omega_{\text{NH Par}} = \sqrt{\frac{c_{\text{Nap v}} + c_{\text{Pneu v}}}{m_{\text{NH}}}} \quad (1)$$

where m_{NH} is the mass of the unsprung masses, $c_{\text{Nap v}} = c_p \cdot i_p + c_{\text{VT}}$ is the stiffness of the wheel, c_p is the stiffness of main spring, i_p is the ratio between the motion of the wheel and that of the spring and c_{VT} represent the stiffness of other flexible elements. From the comparison of the values of the eigenfrequencies $\omega_{\text{NH Par}}$ obtained by calculation, simulation and experiment it is obvious that the virtual model has reliable properties for further observations (see Tab. 1 and, analogously, Tab.2).

Table 1: Comparison of the eigenfrequencies of the unsprung masses

Axle	Calculation [Hz]	ADAMS/Car [Hz]	Experiment [Hz]
front	11.42	11.5	12.05

rear	13.44	13.4	14.65
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Table 2: Comparison of the eigenfrequencies of the bodywork (sprung masses) in the vertical direction.

Bodywork	Calculation [Hz]	ADAMS/Car [Hz]	Experiment [Hz]
vertical	1.49	1.5	1.68

4 THE INFLUENCE OF THE FLEXIBLE PARTS OF THE UNDERCARRIAGE AND THE BODYWORK

The virtual model of the vehicle is more accurate when it has flexible parts. Using the module ADAMS/Flex, the MNF file which represents the FEA model of flexible parts can be imported into the ADAMS/Car environment. According to the Hurty-Craig-Bampton method of modal synthesis

$$\mathbf{u} = \Phi \mathbf{q} \quad (2)$$

The modal matrix Φ reduces the large number of physical coordinates $\mathbf{u}$ to a smaller number of modal coordinates $\mathbf{q}$ which enables the achievement of an acceptable level of accuracy and computational efficiency in the dynamic simulations. In Tab. 3 there is a comparison of the eigenfrequencies of the FEM and MBS models of the chassis subframe before and after modal reduction.

Table 3: Comparison of the eigenfrequencies of the FEM and MBS models of the chassis subframe.

Eigenfrequency	FEM model [Hz]	MBS model [Hz]	Difference [%]
7.	126.77	126.78	0.0
8.	209.13	209.02	-0.1

The flexible properties of the chassis subframe of the rear axle mostly influence the toe angle, the camber angle and the slip angle of the wheel [5].

Figure 4: Simulation on the four-post test rig of howa vehicle with flexible bodywork crosses bumps.

Figure 5: The course of the PSD of force transfer from the upper mount of the dumper into the bodywork; the upper curve represents rigid bodywork, and the lower is for flexible bodywork.

The Pacejka tire model was used in conjunction with the full vehicle model with flexible bodywork shown in Figure 4 to simulate crossing bumps at a speed of 100 km/h on a four-post test rig. The solver in the ADAMS/Car program was run in the standard mode (with integrator GSTIFF, a sampling frequency of 3000 Hz, required accuracy of the results 0.01, and adaptivity 0).

From the results of the simulations on the four-post test rig (Fig. 4) shown in Fig. 5, it can be observed that the torsional, lateral and longitudinal bending eigenshapes of the flexible bodywork manifest themselves in the frequency range above 30 Hz, which is important for the assessment of durability and noise emissions [6]. Then it can be concluded that, for the purposes of research on the driving dynamics of a vehicle, the virtual model of the vehicle can have rigid bodywork.

3 CONCLUSION

The goal of this research was to develop a methodology to enable the determination of the degree of influence of individual elements, such as flexible mounts, shock absorbers [7], bumpers, suspension arms, the subframe and the bodywork, on the resulting dynamic properties of the vehicle. The full vehicle virtual model, created according to the developed methodology, should have properties corresponding to the results of the actual measurements. This was achieved mainly by applying the methods of reduction and optimization of the static and dynamic characteristics of the individual elements. Also, the flexibility of the arms in the wheel suspensions, the chassis subframes and the bodywork was taken into account. The correctness of the assumptions made based on the theory and modeling approaches was confirmed by the acceptable consistency of the results obtained by calculation and simulation with the actual measurements. Therefore, such a model can be used for the rapid and reliable development of a vehicle with the required characteristics in terms of driving dynamics, namely handling, driving safety, ride comfort, noise emissions, vibration and the harshness of the operational functions (NVH analyses), as well as for the reliable determination of the course of the dynamic load over time.

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