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ANALYSIS OF VIBRATIONS OF THE SIMPLIFIED MODEL OF THE SUSPENSION SYSTEM WITH A DOUBLE SPRING AND A FLUID DAMPER

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Abstract

The subject of the paper are forced vibrations of the rear suspension of the biaxial vehicle fitted with the double spring spatial shell model and the viscous damper, under the force pulse input with the given transient response. The play between the master and auxiliary spring is a specific feature of the double spring design [1, 6]. Numerical tests were taken for three variants of the viscous attenuation. The attenuation coefficient values for typical dependent suspensions and their range of changes were assumed on the basis of publication [14]. The problem of modelling and analysing, including contact issues, was solved with the use of the MSC Software programme package with the special use of MSC Patran pre-processor [12]. The numerical analysis of the suspension model with the geometrical non-linearities, viscous attenuation, and time-varying load was taken with the use of the finite element method (FEM). The selected results of the model tests in the form of the suspension relative displacements (speeds) and deformations (stresses) for the selected points of the master and auxiliary spring against time are presented in the paper. The complete model of the suspension system makes possible to examine the influence of various inputs on the interoperation and the phenomena associated with the dynamic response of such a complex system.

Keywords: *FEM, numerical models, double multi-leaf spring, suspension of a motor truck, viscous damper*

1. Introduction

The computing technique development, including common accessibility to the advanced computing systems, generates new possibilities of assisting design engineering of the mechanical vehicle suspension systems. The implementation of the FEM method to the non-linear geometrically numerical analysis of the rear suspension depending of the truck (delivery van) families of 3.5 t total mass (Fig. 1) is presented in this paper. The suspension system was fitted with a double spring and a fluid damper which operated as a linear viscous attenuator in the suspension model. The spatial model of the elastic element made of the shell components, described in publications [6, 7] was applied. The model combines merits of the spatial solid models and the simplest beam models. Its structure is simple. It generates low „numerical costs" associated with the workload for modifications and provides short analysis times within the range of non-linearity. As a result, the suspension model, with the elastic element bilinear characteristics and the linear attenuation characteristics (Fig. 2), was obtained. Besides, in the suspension model, with the purpose of cognition and interpreting, the sprung mass was not taken into consideration, and the unsprung mass corresponds to the spring mass only. In the further works the model mentioned above will be supplemented and, in its structure, it will include the main counterparts of the real rear suspension used in the given vehicle class. The play (Fig. 1, 2) between the master and auxiliary spring is a specific feature of the double spring [1, 6]. The simulation tests were made for three attenuation variants, under the force pulse input with the given transient response (Fig. 3). The attenuation coefficient values for typical dependent suspensions and their range of changes were assumed on the basis of publication [14]. The researches presented in this paper are the continuation of the double spring strength analysis, described in the papers of the MUT research workers [1,2,3,6,7,8]. The presented models and results were obtained thanks to more advanced modelling and analysis techniques including the contact issues possible to implement with the use of the MSC Software engineering programme package [12, 13].

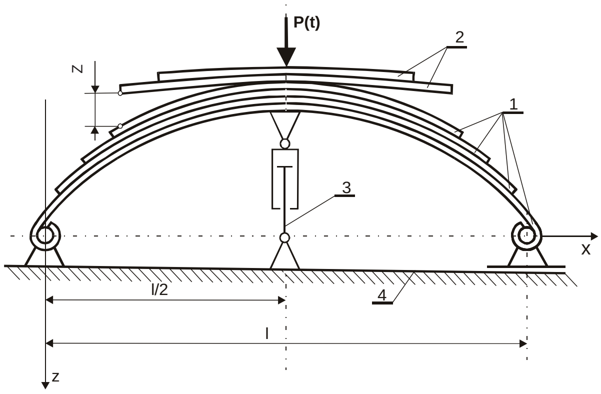

Fig. 1. Scheme of suspension with a double spring and a damper, 1 – master spring leaves, 2 – auxiliary spring leaves, 3 – fluid damper, 4 – frame

2. Numerical FEM Models

The numerical model of the suspension with a double spring and a fluid damper (Fig. 2), and the external load model in the form of the force pulse with the given transient response (Fig. 3) are presented in the paper.

2.1. The Model of the Suspension with a Double Spring and a Damper

The discrete model of the double spring in 3D spatial shell model version, described in details in publication [6], was used for the research. The plays between the master and auxiliary spring (Fig. 1), resulting from the geometrical and technological properties of the tested spring, were being modelled by placing the two-node GAP elements between the nodes of the neighbouring leaves. The interaction conditions for the neighbouring leaves at the two constituent springs was mapped, likewise for the beam model, with the use of the special kinematic MPC elements. It was assumed that the vertical linear displacements of the neighbouring leave nodes, connected by kinematic dependencies, were identical. The linear elastic characteristics of the material was used in the model. The characteristics was described with the following parameters: E=206000 MPa, ν=0.3, ρ=7.8*10^{-6} kg/mm^3. The suspension model was complemented with the simplified model of the viscous damper. DAMPER two-node element with a linear attenuation characteristics was used. The characteristics was described with C attenuation coefficient. The element was placed between an additional node connected with the MPC kinematic elements with nodes on the symmetry plane of the master spring lower leaf and the clamping-to-vehicle-structure node (Fig. 2). The models were generated with the use of MSC Patran pre-processor [12].

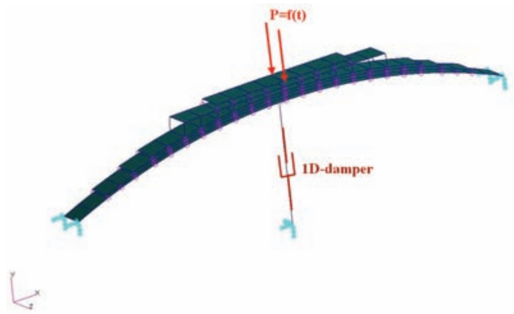

Fig. 2. Model of the suspension with a double spring and a fluid damper

2.1. External Load Model

Two identical concentrated forces make the suspension load. The time-varying forces are applied on the symmetry plane to the auxiliary spring upper leaf surface, as shown in Fig. 2. The force values change from 0 to 5.0 kN within 0 – 0.35 s period according to the curve shown in Fig. 3. The resultant load maximum value occurs after 0.1 s and amounts 10 kN.

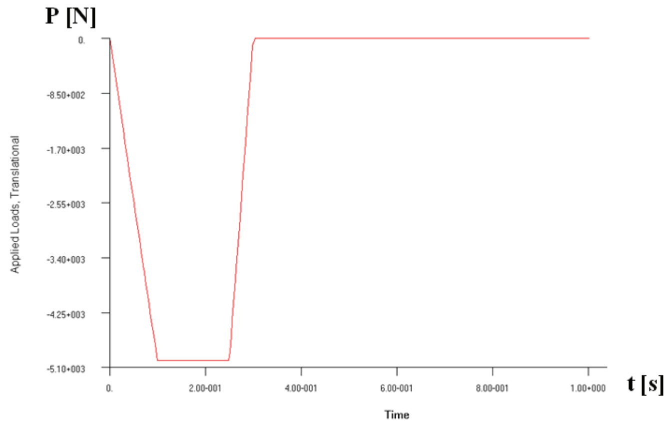

Fig. 3. External load model (P(t) — the force pulse) for the suspension model against time

3. Numerical Analysis — Results

The numerical analysis of the suspension including a double spring (3D spatial shell model) and a viscous damper was made with the use of MSC Nastran software for the *non-linear transient response* [13]. The boundary conditions corresponding to the spring mounting within the vehicle suspension are shown in Fig. 1. For the model nodes corresponding to the outermost edges of the longest leaf the possibility to move along Y and Z axes was blocked.
The suspension numerical model was loaded with two vertical forces 0.5 P(t) = 5 kN. The numerical analysis made with the use of the time-varying loads makes it possible to asses the influence of the forces on the double spring vibrations and interoperation of the master and auxiliary spring leaves including the viscous attenuation. The basic parameters describing the models used in the three variant simulation tests are presented in Table 1. In variant I the vibration system of the suppression model was examined with omission of the damper attenuation. The variant II and III models of the numerical analysis differed in the damper attenuation coefficient only.

Tab. 1. Main parameters of models used in numerical simulation

Parameter	**Variant I**	**Variant II**	**Variant III**
Time of simulation [s]	1.0	0.6	0.6
Loading time [s]	0.35	0.35	0.35
Mass/weight coefficient [s^2/mm]	0.001	0.001	0.001
Damping coefficient of the shock-absorber [Ns/mm]	0.0	1.5	15.0
Structural damping coefficient [Ns/mm]	0.06	0.06	0.06

The node displacements and speeds, as the dynamic quantities of the tested system, were being recorded during the numerical tests as well as H-M-N reduced stresses describing the spring leaf effort. The charts of the node vertical displacement changes under the load force on the symmetry plane of the auxiliary spring upper leaf against time are shown in Figures 4, 6, 9. The charts correspond to the results for I, II, III analysis variants. Similarly, in Figures 5, 7, 10 H-M-H reduced stress changes against time are shown. The stresses were determined in the area of the model nodes neighbouring on the symmetry plane of the upper leaf, and correspond to the results obtained for the individual numerical analysis variants. The node speed vertical component changes on the symmetry plane of the auxiliary spring upper leaf against time, determined for the analysis variant II and III respectively, are presented in Figures 8 and 11.

Variant I — the Vibration System without Attenuation

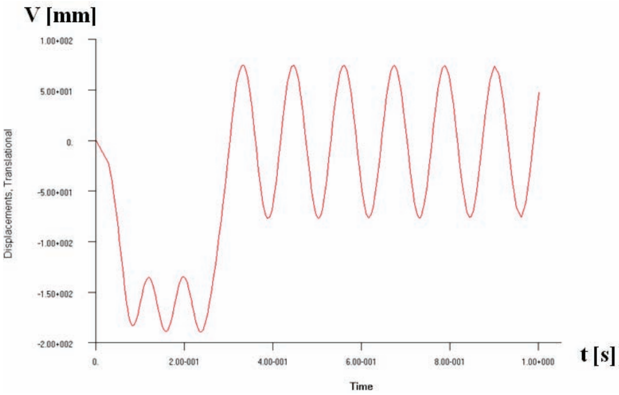

Fig. 4. Node vertical displacements on the auxiliary spring upper leaf symmetry plane

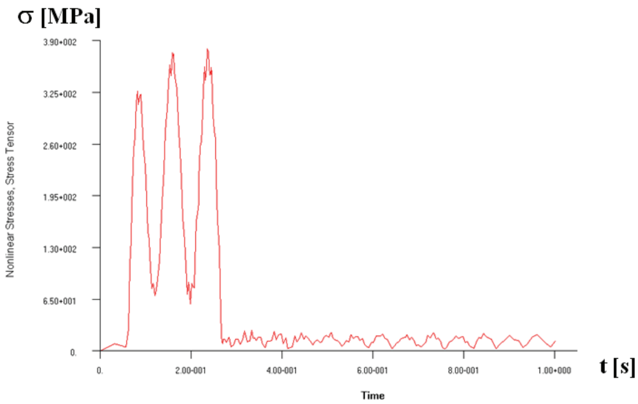

Fig. 5. H-M-H reduced stresses change for the central element of the upper leaf

Variant II — the System with Minimum Attenuation

(the attenuation coefficient of the damper C=1.5 Ns/mm)

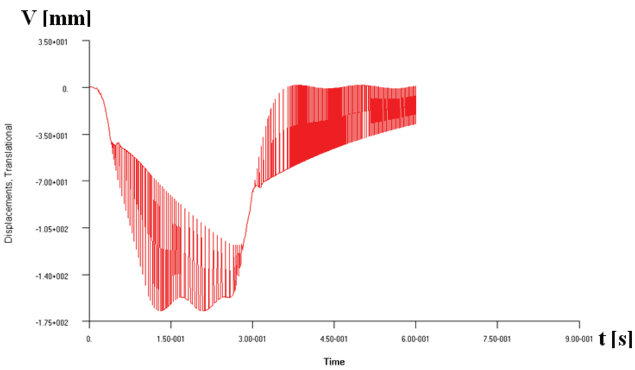

Fig. 6. Node vertical displacements on the auxiliary spring upper leaf symmetry plane

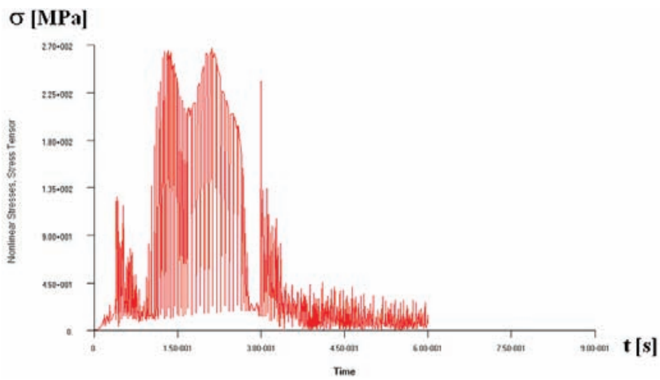

Fig. 7. H-M-H reduced stresses change for the central element of the upper leaf

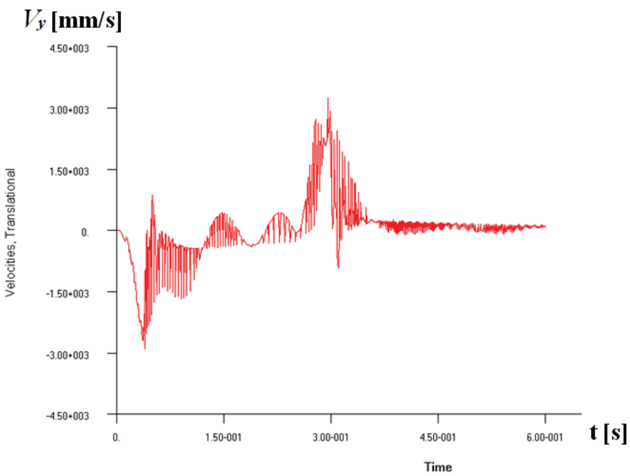

Fig. 8. Node speed vertical component changes on the symmetry plane of the auxiliary spring upper leaf

Variant III — the System with Maximum Attenuation

(the attenuation coefficient of the damper C=15.0 Ns/mm)

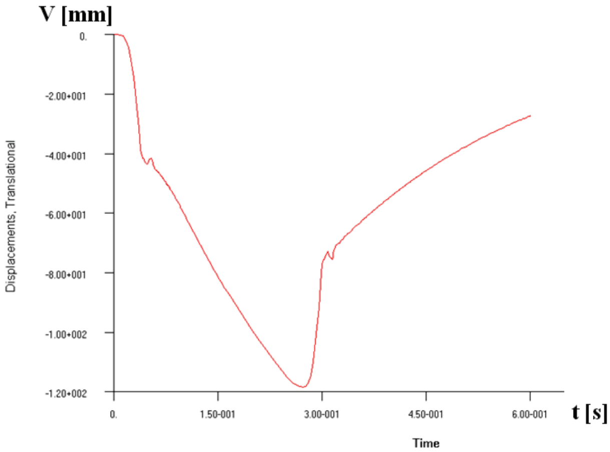

Fig. 9. Node vertical displacements on the auxiliary spring upper leaf symmetry plane

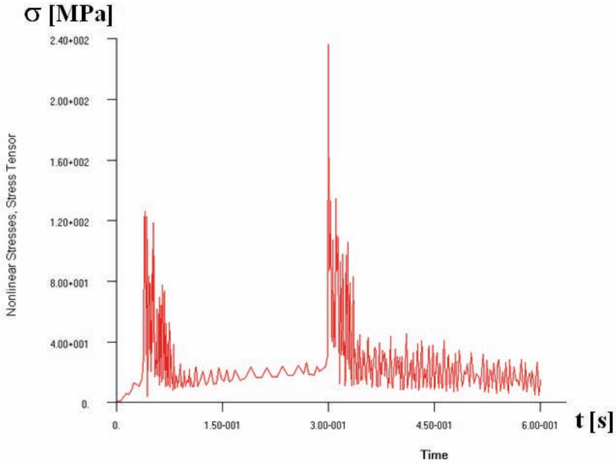

Fig. 10. H-M-H reduced stresses change for the central element of the upper leaf

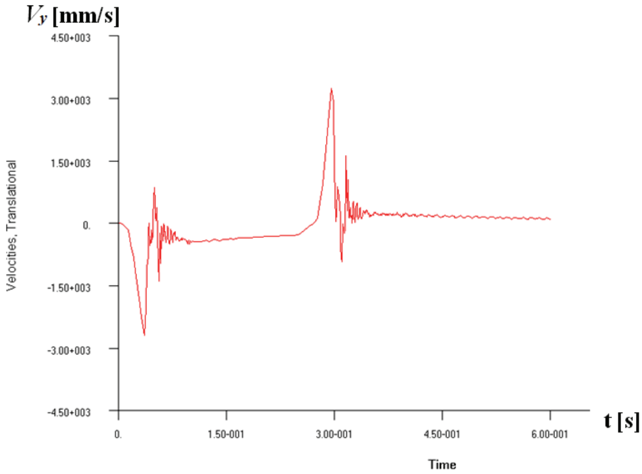

Fig. 11. Node speed vertical component changes on the symmetry plane of the auxiliary spring upper leaf

The displacement maximum values determined for the discussed simulation tests variants were obtained in the load holding phase (before 0.25 s of the analysis time is out) directly before the system unloading phase and during the models unloading (Fig. 3). The maximum values of the node vertical displacement on the auxiliary spring upper leaf symmetry plane, maximum reduced stresses determined in the area of the model elements neighbouring on the upper leaf symmetry plane, and maximum speeds determined for the individual analysis variants are presented in Table 2.

Tab. 2. A comparison of the selected results of dynamics analysis

Parameter	**Variant I**	**Variant II**	**Variant III**
Maximal vertical displacement **V [mm]**	190	170	117
Maximal H-M-H stress **σ [MPa]**	380	265	235
Maximal vertical speed **V_y [m/s]**	-	2.95/3.3	2.7/3.3

The are two speed maximum values: the first one recorded during the input acting and the second one recorded during the unloading final phase. The greatest system vertical displacements and maximum stress values were recorded for variant I, when the damper was not taken into account. For the models corresponding to the numerical analysis variants II and III all discussed quantities decrease along with the increase of external attenuation defined for the damper model.

4. Summary

The double multi-leaf spring (with a bilinear rigidity characteristics), usually used in the rear dependent suspensions of the trucks and delivery vans, was the subject of the research described in this paper. The problem of interoperating of the spring with the damper simplified model was discussed. In order to examine the action of the system mentioned above more precisely the FEM shell model of the double multi-leaf spring was modified. Next the numerical analyses of the discussed suspension elements were made with the use of MSC Nastran programme. The following boundary conditions were taken into account: asymmetrical mounting with the possibility of the spring single-side slip along the horizontal bearing surface and loading with the force pulse input linearly variable against time. Applying the precise model of the spring in the suspension system makes it possible to test deformations, the leaf interoperation results and the efforts of the individual double spring leaves during various phases of the input acting. The complete model of the suspension system with the double springs will make it possible to examine the influence of various inputs on the interoperation of the individual system elements, and the phenomena associated with the dynamic response of such a complex system under the spatial loads acting.

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