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NUMERICAL STUDIES OF SUSPENSION SYSTEM WITH DOUBLE SPRING LOADED USING THE FORCE PULSE

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Abstract

The forced vibration of a dependent rear suspension model of the two-axle vehicle, equipped with a 3D model of a double spring and a viscous shock absorber is the subject of consideration. The clearance between the master and auxiliary springs is a specificity of the double suspension construction. Dynamic analysis of such a system has been performed by forcing with different forms of force pulses using the fixed base of force 10 kN. Three different time runs (two trapezoidal and one triangular model) are used as the force impulses. Modelling and analysis taking into consideration contact problems have been developed using the MSC.Software package with a special use of Patran pre-processor and Nastran solver. A finite element method (FEM) has been used in numerical analysis of the suspension model with geometric nonlinearities, viscous damping [9] and time-varying load. The boundary conditions of numerical models correspond to a spring support in the real vehicle suspension. The non-periodic damped vibrations and damped free vibrations of the simplified suspension system are investigated during numerical analysis. Selected results of the model tests have been presented in the form of the timings of the accelerations of the spring elements and of the forces in the suspension damper.

Keywords*: dependent motor truck suspension, double multi-leaf spring, viscous damper, numerical analysis, FEM*

1. Introduction

The development of computational techniques, including the general availability of advanced computational systems, creates new possibilities for design of a suspension of motor vehicles [1-3, 10, 11, 12-15]. The use of a finite element method (FEM) [2-8] for the geometrically nonlinear numerical analysis of the dependent rear suspension of a family of trucks (commercial vehicles) with a total weight of approximately 3.5 tons is presented in the article. A numerical model of the rear suspension is equipped with a double spring and a hydraulic shock absorber (Fig. 1) which interacts as a linear viscous damper in the model. Numerical studies of suspensions are the result of continuation of work [5-8] at Military University of Technology (MUT) on the prototype multi leaf double spring compound of the master and auxiliary springs. The auxiliary spring is switched to full operation when a load of the system exceeds the critical value. As a result, a spring stiffness curve is bilinear. The numerical analysis of the double spring using a damper in the form of a special element with linear reaction is discussed in the article. The system is loaded by the force impulses with equal amplitude and three different time runs. Such time-varying loads will allow assessing an influence of the forces on the double spring oscillations and the interaction of the master and auxiliary springs with respect to the viscous damping.

2. The forced vibration of the suspension system with damping

Numerical analysis of a suspension compound of a double spring (3D shell model) and the viscous shock absorber (Fig. 1) has been performed using the nonlinear dynamic procedures in

MSC. Nastran code [4, 10, 11, 12]. A response of the suspension system to the time-varying force excitation is determined by integration of the motion equation system (1) in the Transient Response Analysis [11]. Equations (1) include damping of the system (2) and the external load variables over time *P*(t) [11].

$$[B]\{\ddot{q}\}+[C]\{\dot{q}\}+[K]\{q\}=\{P(t)\}, \tag{1}$$

where:

$[C]$ – velocity-related damping matrix,
$[K]$ – stiffness matrix,
$[B]$ – inertia matrix,
$\{P\}$ – vector of external load,
$\{\ddot{q}\}$, $\{\dot{q}\}$, $\{q\}$ – vectors of acceleration, velocity and displacement.

The damping matrix C of the test system is determined according to relationship (2) [11]

$$[C]=[C^1]+[C^2]+\frac{G}{W3}[K], \tag{2}$$

where:

[C^1, C^2**]** – matrices of element damping and transformation functions,
G – structural damping factor,
$W3$ – vibration frequency presented in radians per unit of time,
$[K]$ – stiffness matrix of the system.

Integration of motion equations (1) is carried out according to a differential method algorithm [11]. Velocity and acceleration are expressed in discrete form (3). The motion equations integrated by the differential procedures are written using formula (4).

$$\{\dot{q}_n\}=\frac{1}{2\Delta t}\{q_{n+1}-q_{n-1}\} \quad , \quad \{\ddot{q}_n\}=\frac{1}{\Delta t^2}\{q_{n+1}-2q_n+q_{n-1}\}, \tag{3}$$

$$\left[\frac{B}{\Delta t^2}\right](q_{n+1}-2q_n+q_{n-1})+\left[\frac{C}{2\Delta t}\right](q_{n+1}-q_{n-1})+\left[\frac{K}{3}\right](q_{n+1}+q_n+q_{n-1})= \\ =\frac{1}{3}(P_{n+1}+P_n+P_{n-1}), \tag{4}$$

where:

Δt – time step,
q_{n-1}, q_n, q_{n+1} – displacements in steps respectively: (*n*-1), (*n*) and (*n*+1).

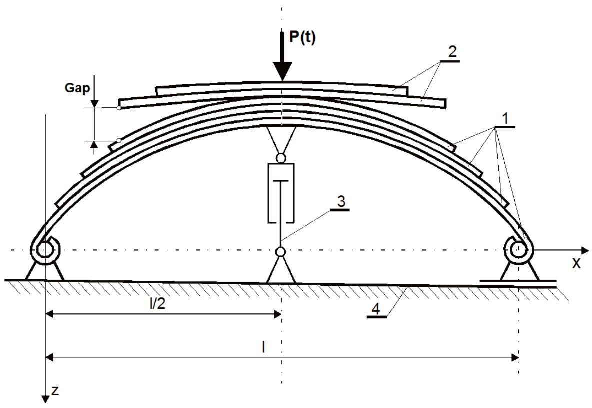
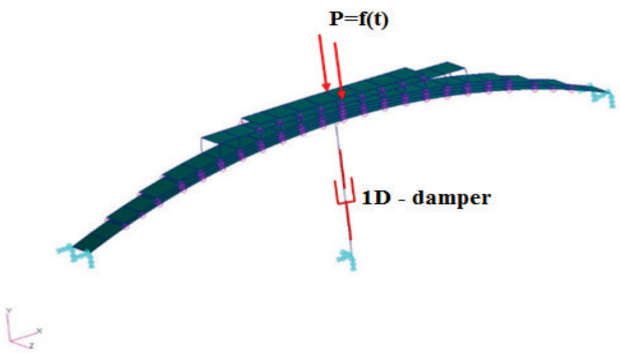


Fig. 1. Diagram of the dependent rear suspension equipped with a free double spring and a hydraulic damper 1 – master spring; 2 – auxiliary spring; 3 – hydraulic damper; 4 – longitudinal component of the chassis frame [12]

A 3D shell model of the double spring (Fig. 2) has been used for numerical tests [4-8, 12]. Clearances between the leaves of the master and auxiliary springs (Fig. 1), resulting from the geometrical and technological features of the tested spring, have been modelled by introducing "node to node" contact – GAP elements between two nodes of adjacent leaves [10, 11]. The conditions for interactions of the leaf ends with adjacent leaves in both component springs have been mapped using MPC kinematics elements [10-12]. It has been assumed that linear displacements in the vertical direction of neighbouring nodes associated with kinematic equations are the same. The suspension model has been completed with a simplified model of a viscous shock absorber. A two-node DAMPER element with a linear damping characteristic described by a damping factor C has been used. The magnitude of the damping factor for typical dependent suspensions, its range of changes has been adopted from the article [8].

a)



b)

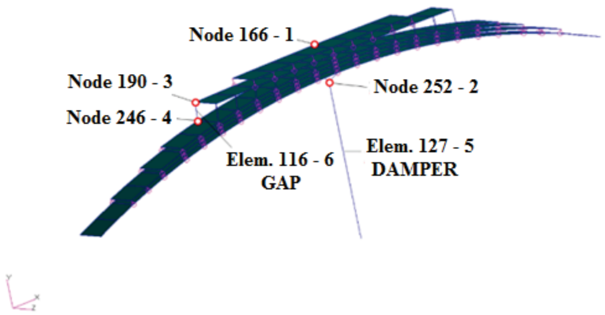


Fig. 2. FE model of the dependent suspension equipped with a free double spring and a hydraulic damper: a) FE model with boundary conditions and load model [12]; b) FE model with description of nodes 166, 190, 252, 246 (marked as respectively: 1, 3, 2, 4) and elements (Elem. 116 – 6, Elem. 127 – 5)

The load in the suspension model is reduced to two concentrated forces, variable over time, applied in the symmetry plane to the top surface of the auxiliary spring, as in Figure 2. The value of these forces from 0.0 to 5.0 kN changes in the time interval 0-0.3 s according to the scheme shown in Fig. 3. The maximum value of the resultant load $P = P_{max}$ of the system occurs after 0.1 s and is 10 kN.

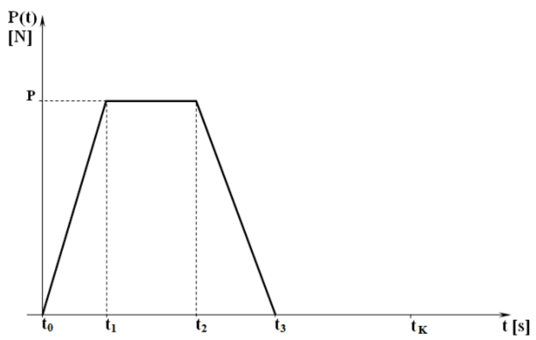


Fig. 3. Model of rear suspension load P(t) – force pulse: t_0 = 0 [s] – start time; t_1 – rise time; t_2 – duration of unchanged force; t_3 – time to reach pulse value equal to 0; t_k – duration of numerical analysis (t_k = 0.6 s)

An impulse of force is described by relation (5):

$$I = \int_{t_0}^{t_3} P(t) \cdot dt \,, \tag{5}$$

The change of the force value over time is expressed by equations (6):

$$P(t) = \begin{cases} P(t) = 0 & \text{for} \quad t_0 = 0\,, \\ P(t) = P\,\dfrac{t}{t_1} & \text{for} \quad 0 \le t \le t_1\,, \\ P(t) = P & \text{for} \quad t_1 \le t \le t_2\,, \\ P(t) = P\left(1 - \dfrac{t - t_2}{t_3 - t_2}\right) & \text{for} \quad t_2 \le t \le t_3\,, \\ P(t) = 0 & \text{for} \quad t \ge t_3\,. \end{cases} \tag{6}$$

Detailed parameters of the force pulse *P*(t) are presented in Table 1.

Tab. 1. Parameters of force pulse P(t) according to Fig. 3

Mark, number, form	Pulse parameters		Pulse duration [s]			
	value [N·m]	duration [s]	t_0	t_1	t_2	t_3
T1 (Trapezoidal)	0.225	0.300	0	0.1	0.250	0.300
T2 (Trapezoidal)	0.150	0.225	0	0.1	0.175	0.225
T3 (Triangular)	0.075	0.150	0	0.1	0.100	0.150

Results of numerical tests

The boundary conditions corresponding to a spring support in the vehicle suspension are shown in Fig. 2b. Linear displacement along the *Y* and *Z* axes is blocked in the model nodes corresponding to the terminal edges of the longest leaf. Numerical analysis performed using variable loads over time allows assessing an influence of forces on the double spring oscillations and the interaction of the master and auxiliary spring leaves with respect to the viscous damping. Changes of the vertical acceleration of node 1 (Fig. 2b) in the symmetry plane of the top leaf of the auxiliary spring with the system damping variant *C* = 15 N·s/mm for the load pulse T1 are shown in Fig. 4. Changes of the dumper forces versus time in variants with different load pulses T1, T2, T3 are shown in Figure 5-7.

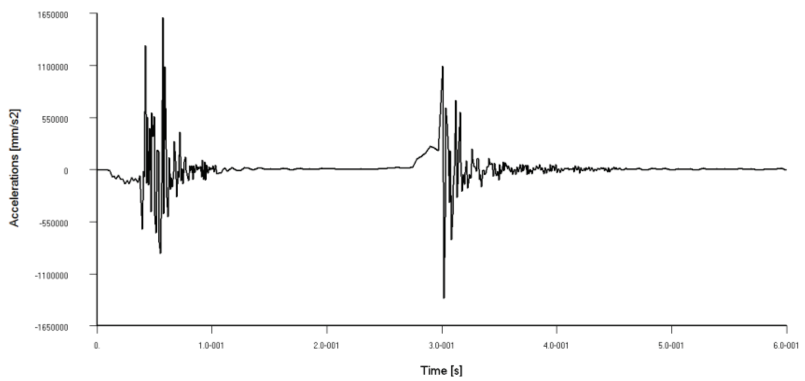


Fig. 4. Vertical acceleration – time trace (a_y max = mm / s^2) determined in element 166 – node 1 (see Figure 2b) for pulse T1

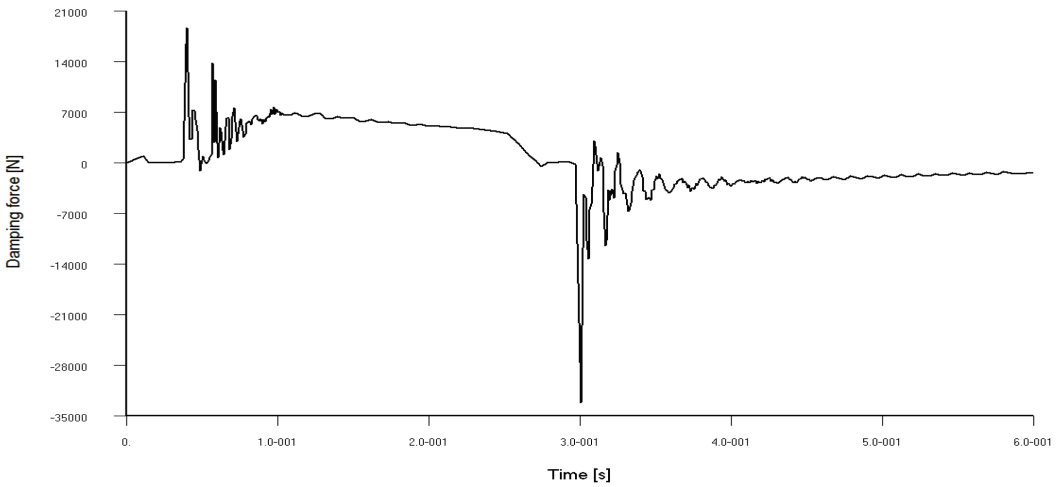


Fig. 5. Dumper forces ($F_{y\,max}$) – time trace for load pulse T1

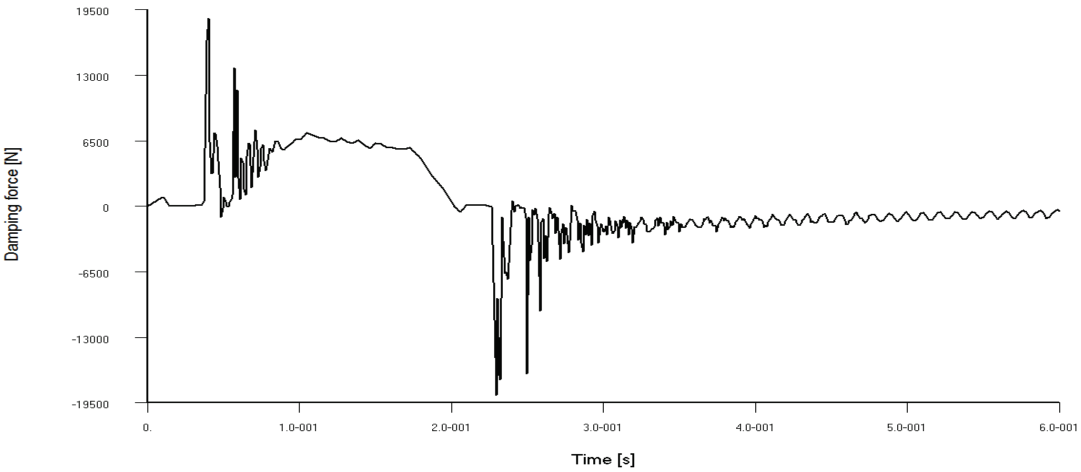


Fig. 6. Dumper forces ($F_{y\,max}$) – time trace for load pulse T2

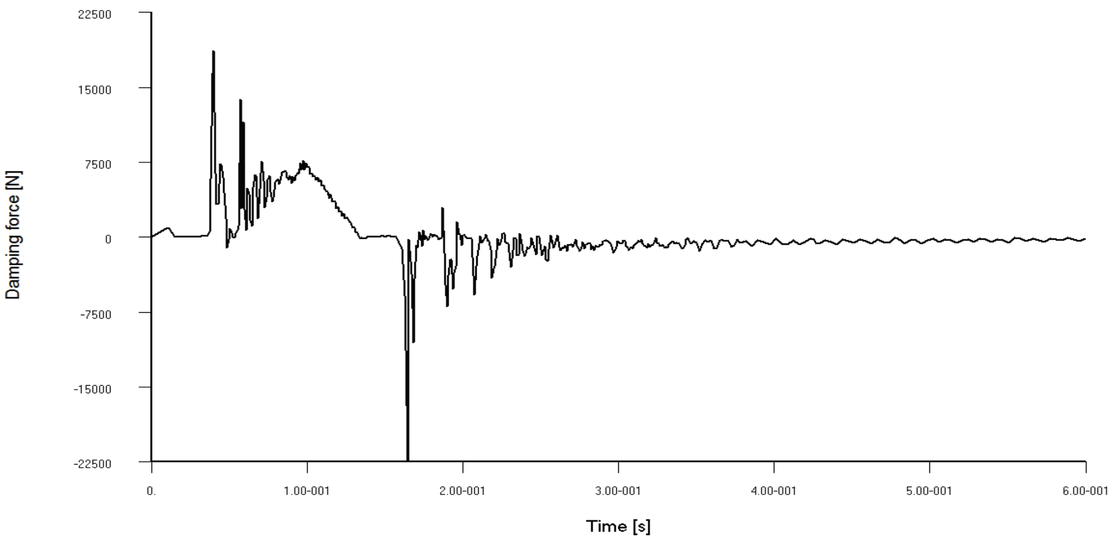


Fig. 7. Dumper forces ($F_{y\,max}$) – time trace for load pulse T3

The suspension system characteristics $F_d = f(V_y)$, in terms of the shock absorber damping force as a function of the speed deformation of the auxiliary spring measured at node 1 (see Figure 2b) for T1 load pulse variant (see Figure 3), are shown in Figs. 8-11. The diagrams represent the characteristics for the subsequent pulse durations from t_0 to t_k (Fig. 3). The complete characteristic $F_d = f\,(V_y)$ for the pulse duration from $t_0 = 0$s to $t_k = 0.6$ s is presented in Fig. 8.

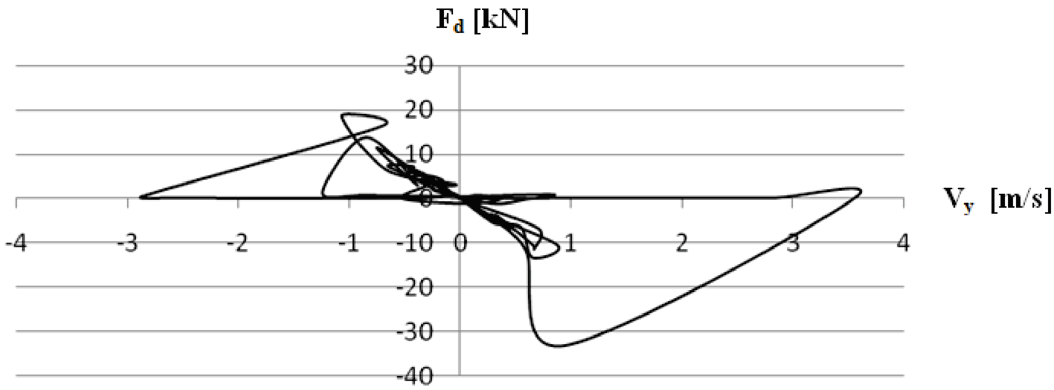


Fig. 8. Curve of the damping force as a function of speed deformation of the auxiliary spring (node no. 1 – Fig. 2b) $F_d = f\,(V_y)$ for pulse T1 and duration from $t_0 = 0s$ to $t_k = 0.6\ s$ – the complementary characteristics of suspension system

The graphs in Figure 9-10 correspond to forced vibrations of the system in the time interval (T1) from t_0 to t_3. Changes of dynamic force $F_d = f\,(v_y)$ as a function of relative velocity v_y for the interval time: t_0-t_1 = 0-0.1 s are presented in Fig. 9, t_1-t_2 = 0.1-0.25 s in Fig. 10a and t_2-t_3 = 0.25-0.3 s in Fig. 10b. The characteristic presented in Fig. 11 shows the free oscillations of the considered system after the forced impulse ends (for $t > t_3$ or t_3-t_k).

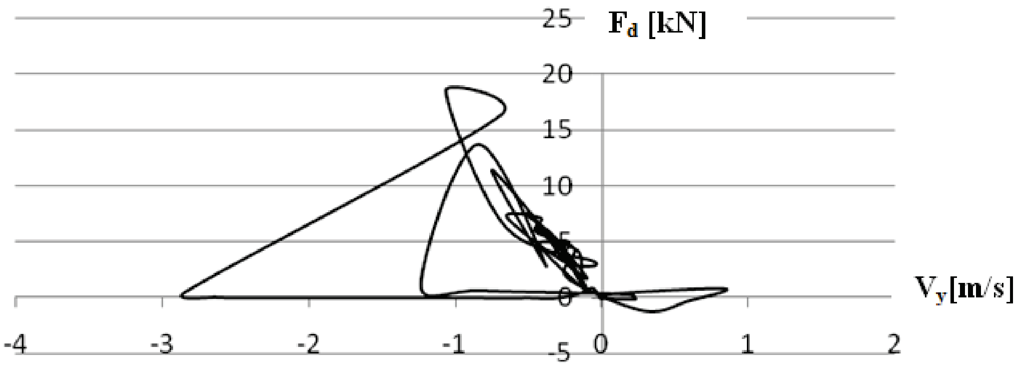


Fig. 9. Partial characteristics of the suspension system in the form of function $F_d = f\,(V_y)$ during the rise time of pulse T1 from $t_0 = 0$ s to $t_1 = 0.1$ s

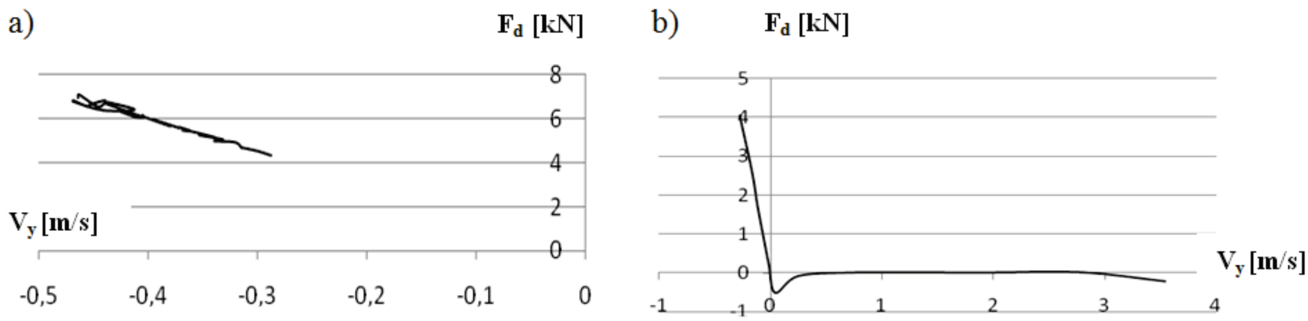


Fig. 10. Partial characteristics of the suspension system: a). in the form of function $F_d = f\,(V_y)$ during the pulse duration T1 from $t_0 = 0.1$ s to $t_1 = 0.25$ s, b). Partial characteristics of the suspension system in the form of function $F_d = f\,(V_y)$ during the pulse reduction T1 from $t_2 = 0.25$ s to $t_3 = 0.3$ s

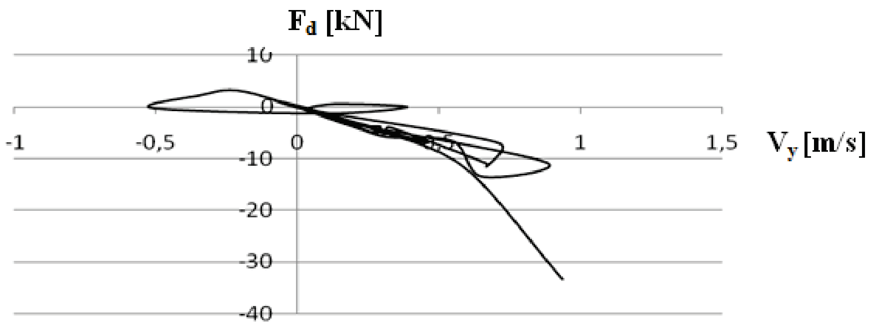


Fig. 11. Partial characteristics of the suspension system in the form of function $F_d = f\,(V_y)$ in the time interval from $t_0 = 0.3$ s to $t_1 = 0.6$ s – complex free oscillations of the vibration system

The maximum changes of the dynamic force ($F_d = |F_{d\,max}$-$F_{d\,min}|$) are recorded in the phase of the increasing load during: t_0-t_1 = 0-0.1 s and after its impact t_3-t_k = 0.25-0.3 s – in Fig. 11. The maximum values of dynamic force changes are ΔF_d (t = 0-0.1 s) < 21 kN and ΔF_d (t = 0.3-0.6 s) < 37 kN. Maximum values of dynamic force change for the multi-leaf spring in the phase of constant load interaction and during decreasing loading, they do not exceed 5 kN are shown in Tab. 2.

Tab. 2. Maximum changes of the dynamic force for the multi-leaf spring in the individual phases of force and during free vibrations

Indication	Vibrations – T1 pulse of force *P*(t)			
	Damped forced vibration			Damped free vibration
Phase of load pulse according to Fig. 3	t_0-t_1	t_1-t_2	t_2-t_3	t_3-t_k
Duration time [s]	0-0.1	0.1-0.25	0.25-0.3	0.3-0.6
Maximum changes of dynamic load ΔF_d [kN]	> 20	> 3	> 5	> 35

Conclusions

The selected elements of dynamical investigations of the simplified suspension system including the bilinear multi leaf spring have been presented in the article.

1. The construction of numerical FE models of the system has been discussed. Models have been verified for static analysis using experimental results from the spring stand tests [6]. Based on the results from the static analysis, it is found that the developed FE models correctly describe the interaction of the master and auxiliary spring leaves, and can be used for further strength tests including dynamic effects.
2. FE model of the rear suspension system of the truck vehicle including a double bilinear multi leaf spring and a hydraulic shock absorber accurately maps the real suspension system. FE model enables the numerical study of the effects of various (discrete and continuous) forces on the interaction of individual suspension elements and phenomena related to the dynamic loads of its components (spring, shock absorber, longitudinal chassis component).
3. Suspension vibrations forced by force impulses (T1, T2, T3) during the rise time of pulse (t_0-t_1) are the non-periodic damped vibrations (Fig. 4-7). Significant vibrations in the suspension system have not been observed in the time interval (t_1-t_2) when the force value reaches the maximum and is constant. The system is re-excited to the non-periodic damped vibrations in the time interval (t_2-t_3) when the force changes from P_{max} to 0. The maximum response amplitude values recorded are higher than during the force rises (t_1-t_2). Oscillations observed for the interval time $t > t_3$ are damped free vibrations.

4. A relationship for the considered suspension system in the form of $F_d = f(V_y)$, where the damping force as a function of the speed deformation of the auxiliary spring varies from (+20) to (-35) kN, when the velocity V_y changes from (-3) to (+3) m / s at the same time interval. Based on the analysis, it has been found that the damping force of a shock absorber reaches the extreme values in the following cases: $F_{d\,max} < +20$ kN for $V_y = -1$ m/s; $F_{d\,min} < -35$ kN for $V_y = +1$ m/s.

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