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Vibrations Simulation of Wheeled Vehicles

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Abstract. The paper deals with vibration analysis for wheeled vehicles and effect of suspension system on ride comfort. Subject of the analysis constitutes a vehicle wheel suspension system designed for special applications. The results of the contribution present an analysis of simulation of different suspension systems for riding over terrain obstacles.
Keywords: mechanics, wheeled vehicle, vibrations, suspension system

1. INTRODUCTION

Investigation of vibrations of wheeled vehicles is a very complex process, which has major influence on overall driving performance and behaviour of the vehicle in the process of design of a suspension system. Vibrations occur mainly when riding on road irregularities or on ground terrain. The vehicle represents a dynamic vibratory system, whose inputs constitute irregularities of the road surface. The output is the system’s feedback which takes for example the form of vertical displacement, velocities, accelerations, etc. which constitute the criterion of driving comfort.

2. EQUATIONS OF OSCILLATORY MOTION

The equations of motion can be derived using Lagrange equations. The resulting equations constitute a system of linear differential equations with the number of equations given by the number of vibratory system's degrees of freedom:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_i}\right)-\frac{\partial L}{\partial q_i}+\frac{\partial D}{\partial \dot{q}_i}=f_i \tag{1}$$

the Lagrangian is given by:

$$L=T-V \tag{2}$$

q_i – generalized coordinates,
T – kinetic energy of the system [J],
V – potential energy of the system [J],
D – damping energy of the system [J].

The right side of the Lagrange equations expresses generalized non-potential forces. The system is excited kinematically and this excitation is included as part of the potential energy on the left side of the equations. We can further derive the equations of motion using the Newton–Euler method. The principle is in the condition of equilibrium of forces and of moments in the direction of individual axes.

Equilibrium conditions are defined as follows:

$$\sum_i F_{ix}=0 \qquad \sum_i F_{iy}=0 \qquad \sum_i F_{iz}=0 \tag{3}$$

$$\sum_i M_{ix}=0 \qquad \sum_i M_{iy}=0 \qquad \sum_i M_{iz}=0 \tag{4}$$

The proposed derivation method is advantageous especially for a spatial model, where the use of Lagrange equations is difficult in regard to the size of the system. The equations of motion derived from the Newton–Euler equations for „linear models“ must be consistent with the equations derived using Lagrange equations. The equations derived using Newton–Euler method are in a form which is valid also for nonlinear characteristics of tires, springs and shock absorbers.

The equations of motion of a linear dynamic model can also be described using matrix notation:

$$\mathbf{M}\ddot{\mathbf{q}}+\mathbf{B}\dot{\mathbf{q}}+\mathbf{K}\mathbf{q}=\mathbf{f} \tag{5}$$

where: **M –** mass matrix,
B – damping matrix,

K – stiffness matrix,

$\mathbf{q}, \dot{\mathbf{q}}, \ddot{\mathbf{q}}$ – displacement, velocity and acceleration vectors,

f – vector of driving forces.

3. SUSPENSION SYSTEMS OF WHEELED VEHICLES

Due to the high demands being placed currently on the suspension systems for wheeled vehicles in terms of ride comfort and driving safety in particular, which is a problem of conventional passive suspension systems, various controllable elements are introduced in the suspension to reduce conflict between ride comfort and driving safety. Eigen frequencies of the axle and vehicle body differ from each other about ten times. Therefore, the damping in the system axle-body is always a compromise between ride comfort and driving safety as illustrated in Figure 1.

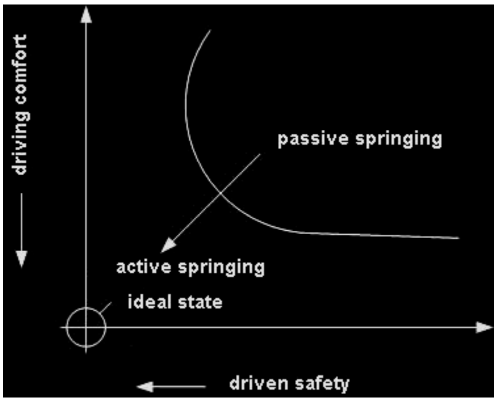

Fig. 1. Conflict between ride comfort and driving safety

The cut-off curve of Figure 1 is the boundary for conventional passive suspension systems of wheeled vehicles. By using damping systems with a variable damping effect, variable properties of vehicle suspension can be achieved, thus getting into the area below the limit curve.

3.1. The distribution of suspension systems of wheeled vehicles

According to principle the suspension systems of wheeled vehicles can be divided (Fig. 2) into:

- passive, Fig. 2a,
- adaptive, Fig. 2b,
- semi-active, Fig. 2c,
- active, Fig. 2d.

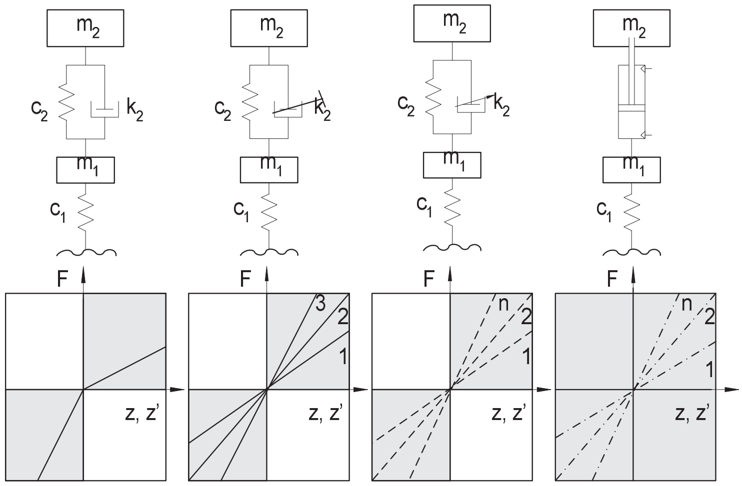

Fig. 2. Two-mass models of suspension systems with damping characteristic:
a) conventional; b) adaptive; c) semi-active; d) active

3.1.1. Conventional suspension systems (Fig. 2a) are systems, which use springs (c_2) and shock absorbers (k_2) of various construction to soften and dampen the reactions from the ground. These systems offer very good construction design at affordable price and are therefore the most prevalent. Such systems however do not change the damping characteristics, which does not conform to the requirement on the vehicle operation.

3.1.2. Adaptive suspension systems (Fig. 2b) are complementary to the conventional suspension system. Shock absorber (k_2) can be adjusted in several stages by step change the damping characteristics. The change in stiffness of the shock absorber can be implemented automatically or selected by the driver. When the change in setting of the shock absorber has already occurred, the system behaves as a conventional suspension system. There may be different soft and hard damping characteristics according to the needs of the driver.

3.1.3. Semi-active suspension systems (Fig. 2c) are also reduced to the function of a shock absorber (k_2), i.e. on the work in the first and third quadrant of the damping characteristics (tension, compression). The working area as opposed to adaptive systems is no longer limited to a few degrees of adjustment, but the change in damping characteristics is progressing smoothly which results into infinitely many degrees of settings (of the characteristics). It is crucial to achieve a high rate (10 ms) of change in the setting.

3.1.4. Active suspension systems (Fig. 2d) work in all four quadrants of the damping characteristics. The action element exerts forces according to relative displacement between the axle (m_1) and vehicle body (m_2). The force between the axle and body is dependent ,besides the relative path and velocity, also on other variables of the system such as derivations, links and frequency functions. The force is usually drawn from hydraulic system consisting of pump, tank, control valves and working cylinders. If such system covers the frequency area only just above the eigenfrequency of the vehicle body, it is referred to as a slow active system. Systems which also cover the area of eigenfrequency of the axle (10-15 Hz) are referred to a fast active systems.

4. SUSPENSION SYSTEM ANALYSIS OF THE ALIGÁTOR 4×4 VEHICLE

4.1. Suspension and springing system of the ALIGÁTOR 4x4 vehicle

The suspension of front wheels and rear wheels is realized by using trapezoidal hinges suspended on coil spring with a hydraulic shock absorber (Fig. 3).

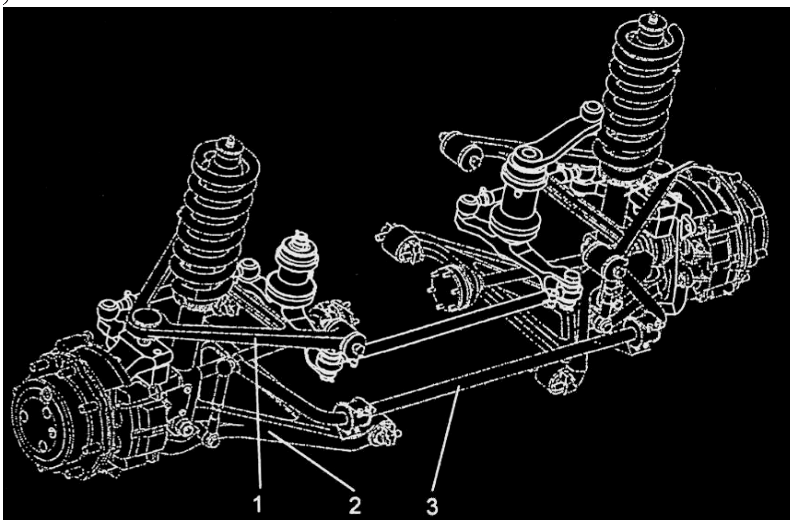

Fig. 3. Suspension and springing of the wheels of the vehicle ALIGÁTOR 4x4

The trapezoidal hinge is solved using a pair of lateral triangular hinge arms. Two lateral arms provide guidance of the wheels and transmission of all forces and moments. If the rotation axes of the hinge arms with regard to the vehicle body are in generally inclined positions, it is a spatial mechanism, whose mobility is assured only under condition that the outer joints, which link the hinged arms with the wheel axle allow spherical movement.

In this case the additional degree of freedom of the wheel movement has to be blocked by another lateral arm, which is in the case of steering wheels, a part of the steering mechanism.

4.2. Experimental measurements

Experimental measurements of the suspension system on the ALIGÁOR 4x4 vehicle were carried out on a track with irregularities of 100 mm in height, across which the vehicle rode at different speeds. The outputs were displacements, velocities and accelerations in the vehicle's centre of gravity. The seats for the operating personnel are positioned around the centre of gravity and they are rigidly mounted to the body of the vehicle with screws. It is therefore assumed that the quantities measured within the centre of gravity would represent actual effects on personnel with sufficient accuracy. Measurements were carried out by a Brüel & Kjaer PULSE system device with following specifications: Type 3560 C with 5/1 channel Input/Output Controller Module Type 7537 and a 12 channel Input Module Type 3038. Triaxial IEPE Piezoelectric Accelerometer Type 4524B was used to collect the data although only quantities in vertical axis are evaluated and presented as part of this paper. The data were sampled at a rate of 100 Hz and evaluated using Pulse LabShop software. The measuring device was placed inside of the vehicle with sensors mounted in the approximate position of the vehicle's centre of gravity (Fig. 4).

Fig. 4. Location of the PULSE system

The rate of discomfort for the operating personnel was evaluated in terms of Root Mean Square (RMS) of vertical acceleration. As the acceleration RMS increases so does the discomfort of the personnel (Tab. 1). In terms of real application the speed has to be limited for different types of terrain and reduced even more to overcome obstacles. The 100 mm obstacle presented here is one which the vehicle is supposed to safely overcome at the speed of 25 km·h^{-1}.

Table 1. Discomfort levels

RMS Acceleration [m·s^{-1}]	Comfort level
≤ 0.315	Not uncomfortable
0.315 to 0.630	A little uncomfortable
0.500 to 1.000	Fairly uncomfortable
0.800 to 1.600	Uncomfortable
1.125 to 2.500	Very uncomfortable
≥ 2.500	Extremely uncomfortable

In Figure 5 the plot of the centre of gravity displacement is depicted when driving across a curbstone 100 mm in height at a speed 25 km·h^{-1}.

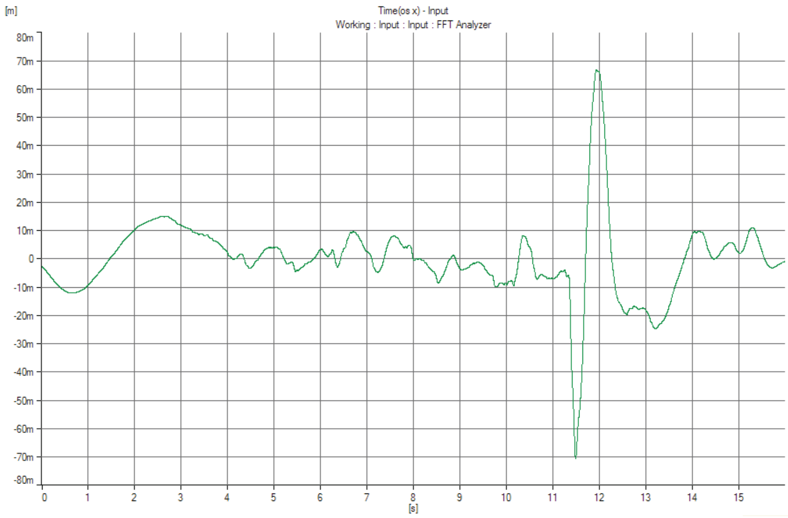

Fig. 5. Plot of the displacement in the centre of gravity of the vehicle

In Figure 6, a plot of the acceleration of the vehicle’s centre of gravity is depicted when driving across a terrain irregularity 100 mm in height at a speed of 25 km·h^{-1}.

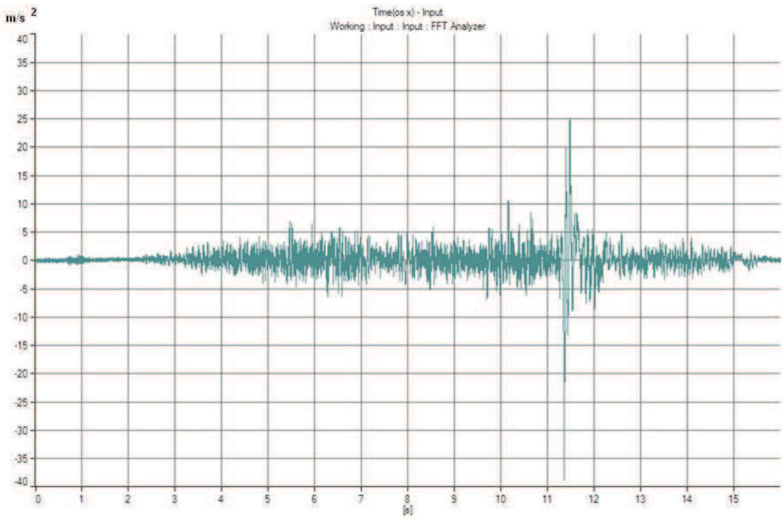

Fig. 6. Plot of the acceleration of the vehicle's centre of gravity

The RMS of vertical acceleration from experimentally obtained data is 5.28 $m \cdot s^{-2}$ which translates into approximately 0.54 G. These accelerations are extremely uncomfortable for the personnel but again this is a military vehicle with specially trained personnel and the speed and obstacle (sharp edged curbstone) presented are designed to simulate an extreme situation such as an evasive manoeuvre made during an ambush.

4.3. Simulation of suspension

The quarter model of the vehicle suspension consists of a two-mass system in Figure 7. Sprung mass m_2 is composed of the vehicle body, passengers and cargo. Unsprung mass m_1 is the mass of the wheel with the axle. The parallel connection of the shock absorber with damping factor k_2 and spring with stiffness c_2 between these two components, creates a passive suspension. The tire is represented by its radial stiffness c_1. Damping characteristic of the tire are neglected because it is few magnitudes less than the radial stiffness of the tire.

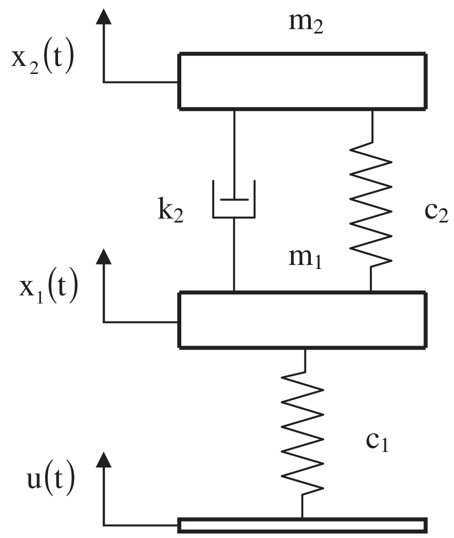

Fig. 7. Two-mass model of wheeled vehicle (spring – shock absorber)

In static position $x_2 = 0$, the spring and shock absorber are compressed due to the weight of the vehicle $c_2 \cdot x_{stat} = m_2 \cdot g$, where x_{stat} represents compression of the spring. This compression compensates for the weight, which is afterwards neglected in the equations. When oscillations are induced, relative movement occurs between the road irregularities and the vehicle body m_2.
Equations of motion for two-mass system are as follows:

$$m_1 \ddot{z}_1 = F_{c2} + F_{k2} - F_{c1} \tag{6}$$

$$m_2 \ddot{z}_2 = -F_{c2} - F_{k2} \tag{7}$$

of which:

$F_{c2} = c_2 (x_2 - x_1)$ – spring force [N], (8)

$F_{k2} = k_2 (\dot{x}_2 - \dot{x}_1)$ – shock absorber force [N], (9)

$F_{c1} = c_1 (x_1 - u)$ – force on the tire [N], (10)

where: m_1 – weight of unsprung masses [kg],
m_2 – weight of sprung masses [kg],
k_2 – shock damping coefficient [N·s /m],
c_1 – radial stiffness coefficient of the tire [N/m],
c_2 – spring stiffness coefficient [N/m],
x_1 – displacement of unsprung masses [m],
x_2 – displacement of sprung masses [m],
u – kinematical excitation [m].

After modifying the equations of motion take on the following form:

$$m_1\ddot{x}_1 + c_1(x_1 - u) - c_2(x_2 - x_1) - k_2(\dot{x}_2 - \dot{x}_1) = 0 \tag{11}$$

$$m_2\ddot{x}_2 + c_2(x_2 - x_1) + k_2(\dot{x}_2 - \dot{x}_1) = 0 \tag{12}$$

The simulation model was created using the Matlab software package using the Simulink tool and from equations (11) and (12).

Figure 8 shows the plots for displacement in the vehicle's centre of gravity for different suspension systems when travelling over an obstacle measuring 100 mm in height.

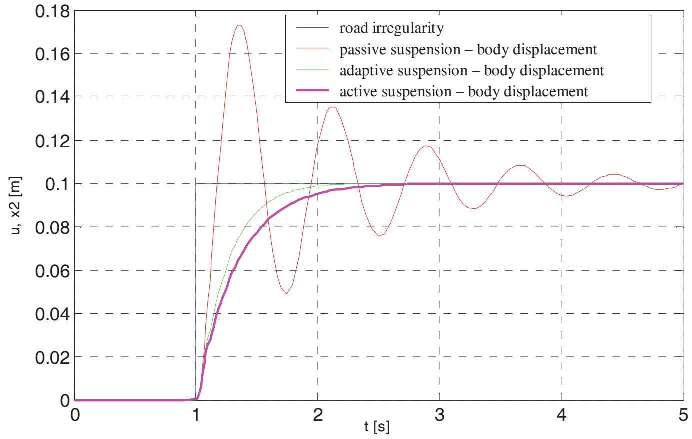

Fig. 8. Plot of the displacement in the vehicle's centre of gravity

The damping coefficient k_2 in (11) and (12) was implemented as a variable depending on the speed of the vehicle and also on the vertical speed and displacement of the vehicle body. The whole set of damping conditions was created gradually to allow smooth ride for a variety of driving conditions and is too complex to be shown. Generally a smaller damping coefficient was favoured for larger vehicle speeds and larger damping coefficient for larger vertical speed of the body which smoothly normalized as the vehicle returned to equilibrium. The result in Figure 8 is of course for optimal conditions and it is a problem to provide optimal damper resistance for all available driving conditions because of limitations in damper displacement or working distance of the vehicle body from the road surface.

Figure 9 presents the plots for acceleration in the centre of gravity of the vehicle for different suspension systems when travelling over an obstacle measuring 100 mm in height.

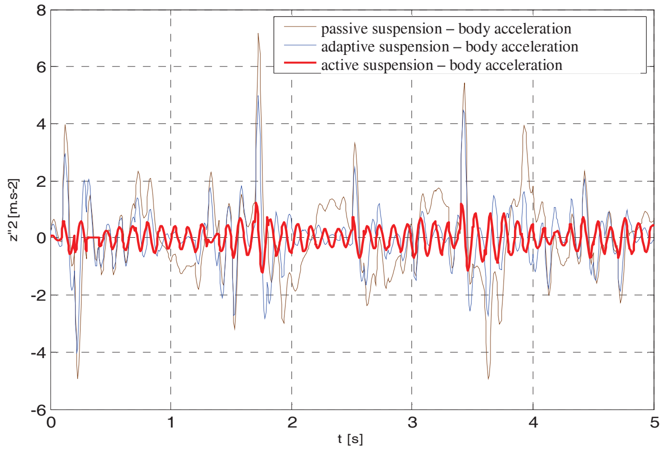

Fig. 9. Plot of the acceleration in the vehicle's centre of gravity

5. CONCLUSION

The paper presents results of experimentally obtained data for vertical acceleration in the centre of gravity of the ALIGATOR 4x4 vehicle at time of driving over a curbstone obstacle to simulate effects on personnel during an extreme situation. The acceleration RMS from the test data reached 5.28 $m \cdot s^{-2}$ which is a value that the personnel perceive as extremely uncomfortable. The vehicle is currently equipped with passive springing thus alternative modes of suspension springing have been evaluated to reduce the discomfort of the personnel. According to the comparison, it is obvious, from point of comfort and safety that response on the obstacle is the most useful when these springing systems are used:

- active springing
- controlled springing
- passive springing.

For the retrieval of relevant results, it is required to analyse and compare each individual kind of springing on the 3D models. The main problem for the active springing is to find optimal set of conditions for the damping force that would cover various driving styles and accommodate for road surface and thus ensuring a comfortable ride. The acceptable vertical displacement of the vehicle body is of course limited and the narrower the displacement range gets, the more strict damping conditions have to be used in order to avoid the use of displacement limiters. This presents a real challenge in the automotive industry with the dilemma being mostly between safety and driving comfort being solved by using preset values for the resistive forces in the suspension, i.e. sport, normal and comfort.

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