Determinations regarding the influence on movement and comfort of different elastic suspension structures in N2 type vehicles

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Abstract. Modern technologies implemented in the automotive field bring a considerable increase of performance, but also safety at the same time. Moving the vehicle under such conditions also requires a high level of comfort, regardless of the type of road traveled. The dynamics and behavior of the vehicle are conditioned by its constructive and functional characteristics. The research carried out highlights some aspects regarding the importance of elastomeric elements in the structures destined for shock and vibration damping. Thus, different solutions adopted are analyzed, for the car’s damping system, to reduce the specific disturbance factors of the road. For a vehicle of average load capacity, respectively 5 tonnes, used in determinations, checks are made on the various elastic systems with role in reducing shocks and vibrations. It is also intended that for the solutions adopted, the frequencies and amplitudes of the vibrations will occur at low values. By modeling the systems in AMESim, the aim is to identify the performance of the solutions adopted, as well as the level of comfort obtained.

1. Introduction

The movement of the vehicles is characterized by the mechanical oscillations produced by the external and internal disturbing factors. The dynamics and behavior of the vehicle are factors conditioned by its technical and functional characteristics. The road and its characteristics determine the dynamics (speed), the safety of movement (longitudinal and transverse stability), as well as the comfort of the vehicle while traveling. The road profile generally causes random disturbances in the structure of the vehicle. The analysis of the different solutions adopted for the components of the suspension system, determines the determination of the optimum level of comfort, for a certain type of vehicle.

The determinations are made on a vehicle with average load capacity (transport) type N2. The profile of the chosen road was the rough one, because it creates a high level of discomfort. The oscillations recorded were of a random type. The research aims to identify an optimal and economically advantageous solution, to reduce the values of vertical acceleration at the level of the vehicle. The identified solution (elastomeric component), represents an additional system, easy to mount, for adapting the suspension to the displacement in rough terrain. The importance of the solution is to reduce the amplitudes of shocks and vibrations, but also to ensure an optimum level of comfort, when traveling the vehicle on unpaved roads.
2. Considerations regarding mechanical oscillations and driving conditions.

The importance of the determinations is the establishment of suitable elastomeric components, in order for adapt the suspensions of the vehicles to traveling in any road conditions. The oscillations recorded at the level of the suspended masses depend on the constants of the elastic system and on the damping of the vehicle. The determinations made are limited only to the vertical movements transmitted by the road and refer to the frequencies: of resonance of the human body (5 Hz), of the vehicles body (0.7-2 Hz) and those transmitted by tires (8-16 Hz). For the human body, the random vibrations generated by the road are very dangerous, if the vertical acceleration and its derivative have high values. Therefore, the accelerations must be observed according to the frequency at which they occur, during the movement of the vehicle. We can say that, the optimal damping is obtained where the resonance amplitude of the system is minimal. International provisions ISO 2631, BS 6408 mention certain values of the vertical acceleration and the physiological influence they have on the human body. Also, the VDI norm 2057 establishes a comfort perception factor (K), calculate by formula (1), depending on the recorded level of vertical acceleration, for frequencies in the range 0.5-80 Hz. [1]

\[
K = \left( \frac{18}{\sqrt{1+\left(\frac{f}{f_0}\right)^2}} \right) \times \sigma_{\ddot{x}}, \quad (1)
\]

where: \( \sigma_{\ddot{x}} \) = the vertical acceleration of the oscillation [m/s²];
\( f \) = frequency [Hz]; \( f_0 \) = reference frequency - 10 Hz.

For objective determinations, it is recommended to sum the squares of the partial indices of comfort of the "n" different variations.

\[
K = \sqrt{\sum_{i=1}^{n} k_i^2}, \quad (2)
\]

where: \( k_i \) = partial comfort index; \( n \) = number of component oscillations.

In Table 1 is presented the scale of subjective perceptions, according to norm VDI 2057.

| Level | Perception threshold (K) | Subjective appreciation of perception |
|-------|--------------------------|--------------------------------------|
| A     | 0.1                      | There is no oscillation              |
| B     | 0.25                     | The first sensations appear          |
| C     | 0.63                     | Minimum level of sensations          |
| D     | 1.6                      | It feels good                        |
| E     | 4                        | He feels strong                      |
| F     | 10                       | Very strong level of perception      |
| G     | 25                       | Very strong level of perception      |
| H     | 63                       | Very strong level of perception      |

For freight vehicles, the difference between the value of the load and the empty vehicle, influences the comfort level. With is the greater this difference, the behavior is the more dynamic, for all frequencies of disruptive accelerations induced by the road.

For create the optimum level of comfort, it is necessary to reduce the body's own vibrations, so that its reducing the amplitude of the acceleration too. This will cause the vibrations to become periodic.
Disruptive pulses (ω) take into account the speed of travel of the vehicle. The irregularities of the road described by function h(t) can be materialized by a continuous distribution of sinusoidal functions, in a varied range of frequencies, with amplitudes and phases that are constantly changing. The amplitude spectra of the disturbing oscillations are in accordance with the accelerations of the car body.

The random function h(t) is described by the relation:

\[ h(t) = \int_{-\infty}^{\infty} \sigma \sin(\omega t + \varphi) d\omega = \int_{-\infty}^{\infty} \sigma_k e^{i\omega t} d\omega, \]  

where: \( \sigma \) = vertical acceleration \( [m/s^2] \); \( \omega \) = pulse \( [s^{-1}] \); \( t \) = time \( [s] \).

The comfort of the suspension system is ensured when the dynamic arrow has a high value. This must be in accordance with the dynamic capabilities of the vehicle, on rough roads. So, for a speeding, on a road with high irregularities, the dynamic arrow \( (\delta_d) \) must be high. The optimum suspension characteristic must provide certain values for the dynamic coefficient and the static arrow. This can be achieved by adopting a system with nonlinear characteristic.

A particular case is the bumps isolated (obstacles, steep holes), which generate excitations in the form of impulses. These are analyzed in the form of step functions.

3. Determinations on the different systems adopted

The analyzes were carried out on an N2 type vehicle, intended for freight and persons transport, but also combined. As such, the static arrow of the suspension changes with the payload carried. For this type of vehicle, the weight of the suspended masses in the loading-unloading process changes, on average, as follows: 10-40% for the suspension of the front axle and 100-150% for the rear axle.

The determinations made do not take into account the characteristics of the tire, the internal pressure and its deformations during the journey, nor the possible deformation of the road (sandy terrain, mud, snow). Knowing the possibilities of loading the vehicle, its destination and the needs of traveling on unpaved roads, it is intended that the suspension system has an optimal capacity to control its overall dynamics. For unpaved roads, the main disturbance and discomfort factor is the vertical oscillation.

In the tests, the observations watch the oscillations at the front axle level of the vehicle. It has load of 3,000 kg. Any changes in the loading of the axles also bring a change in the handling behavior, especially on the rough roads. Also, the unevenness of the road will be felt by the occupants, due to the discomfort created. Three types of elastic systems were adopted at the front axle level:

- simple arm with cylindrical coil spring and hydraulic shock absorber.
- multi-arm with cylindrical helical and hydraulic shock absorber.
- multi-arm with cylindrical coil and magnetorheological shock absorber.

To these systems, the elastic characteristics have been modified by introducing an elastomeric components into the structure. This component (mount), was introducing at the top level of the damping system. The type of elastomer chosen is in accordance with the purpose of its use. The static rigidity of the elastomer is lower than the dynamic one. The relation between the rigidities is a particular one, being dependent on the characteristics of the elastomer. So, static deflection has a higher value and depends the value of the shock or vertical vibration entered the system. There must be sufficient deflection in the system so that the energy of the disruptive oscillation is absorbed by the elastomer. The static deflection of the elastomer is calculated with the relation:

\[ d_{st} = \frac{q_B f_n}{2} = 0,0435 \text{ inch} = 1,104 \text{ [mm]} \]  

The relationship between dynamic (oscillation/shock) and static rigidity is:

\[ K_{sc} \cong 1,4 * K_{st} \]  

where: \( K_{st} \) = static rigidity \( [N/m] \).
3.1. Establishing working conditions and data acquisition

When determining the working characteristics of the elastomer, the vertical acceleration recorded on a rugged terrain was taken into account. The recordings were made with the Vbox system and highlighted with the help of the special program designed by the manufacturer (Race Technology). Its ability to process data, creates the possibility to observe and correlate more details, in connection with the research conducted. In the present case, information was observed on: the planimetry and altitude of the route followed by the vehicle; vertical, horizontal and lateral acceleration; travel speed and distance traveled. In figure 1, data are presented on the vertical acceleration recorded in the rough terrain, but and some data on the movement of the vehicle.

![Vertical acceleration chart](image)

**Figure 1.** Data on the characteristics of the vehicle’s movement

1- Vbox system; 2- Accelerometer IMU 03; 3- Data recorded

The vertical acceleration, at the level of the vehicle body, was recorded with an IMU 03 accelerometer and presented a maximum value of 2.1 g (20.6 m/s²). The speed of the vehicle (38 km/h) and the variation of the altitude of the terrain (max. 928 m) were recorded with the GPS, of the Vbox data acquisition system. The accuracy of the analyzed data is closely related to the sensors and the acquisition system used, as well as to the post-processing operation of the obtained signal.

The processing of vertical acceleration data was performed according to SAE J211, using a Butterworth low-pass filter. Figure 2 shows the block diagram of the acquisition system.

![Block diagram](image)

**Figure 2.** Block diagram of the process of data acquisition and control

The accelerometers used (3) were of digital type, chosen so as to correspond to the purpose of the research. They can operate in three directions, a range of ±16 g, with a maximum 13-bit resolution [2]. Their fixation was made on the unsuspended (spindle) and suspended (body) masses. The acquisition board MEGA 2560 V3 (1), uses an ATmega2560 microcontroller that has 256 KB of flash memory for writing the source code, 8 KB of SRAM and 4 KB of EEPROM [3]. The data is recorded by the Data
Logger (2), which is have an SD port and a clock, working in real time (RTC). The system work program has been written to Flash memory. Data on the compatibility of the various components with the motherboard were also integrated. According to the acquisition procedure, the vertical acceleration is measured with the ADXL 345 accelerometer, processed (filtered) by the MEGA 2560 motherboard and recorded by the Data Logger. The vertical acceleration was pursued due to the major influence it has on the suspension systems of the vehicle, as well as the comfort of the transported persons. Post-processing of data is done with a low-pass filter, using the function [4]:

\[ y(t) = x(t) \cdot \alpha + y(t - 1) \cdot (1 - \alpha) \]  

(6)

where: \( y(t) \) = calculated (estimated) current value; \( y(t-1) \) = calculated previous value; \( x(t) \) = measured value; \( \alpha \) = the attenuation factor (between 0 and 1).

Figure 3 presents the data acquisition equipment about vertical accelerations.

![Figure 3. The data acquisition system about vertical accelerations](image)

1 - MEGA 2560 V3; 2 - Data Logger; 3 - Accelerometer ADXL; 4 - Working directions of the accelerometer

### 2.2. Mathematical calculation of the working characteristics of the elastomeric component

The choice of the type of elastomeric insulator is made taking into account the fact that it must respond to a level of 3 times the equivalent acceleration, specific to the vibration at the desired natural frequency. Knowing the reference frequencies in which the resonance is manifested, in the case of moving vehicles and the human body, for the elastomer is pursued an frequency of 15 Hz and a maximum transmissibility value of 2.0.

Determination of dynamic and static rigidity:

\[
K_v = \frac{(f_n)^2(M)}{9,8} = \frac{15^2 \cdot 800}{9,8} = 18.348,6 \quad \text{[N/m]} \quad (7)
\]

\[
K_{st} = K_v \cdot 1,4 = 40493,1 \cdot 1,4 = 9.174,3 \quad \text{[N/m]} \quad (8)
\]

where: \( K'_v \) = total dynamic rigidity at the specified vibration; \( f_n \) = the natural frequency of the selected system (Hz); \( M \) = weight of the insulated equipment (Kg).

In the case of rough terrain, the vibrations are random, so the analyzes are performed on the basis of probability theory. Thus, the RMS response of the elastomer, to the vertical acceleration type sigma one (\( \sigma_1 \)) is:
\[ g_{\text{RMS}} = \sqrt{\left(\frac{n}{2}\right) \cdot S_i \cdot f_n \cdot T_r} = 14.41 \text{ m/s}^2 \]  
(9)

where: \( g_{\text{RMS}} \) = response to level 1 acceleration;  
\( S_i \) = input random vibration (m/s\(^2\)/Hz);  
\( T_r \) = allowable resonant transmissibility;  
\( f_n \) = the desired natural frequency (Hz).

It is known that elastomers respond to three levels of vibration, so then are calculated:

\[ g_{3\sigma} = 3 \sqrt{\left(\frac{n}{2}\right) \cdot S_i \cdot f_n \cdot T_r} = 5.92 \text{ m/s}^2. \]  
(10)

where: \( g_{3\sigma} \) = the response to level 3 acceleration.

The movement developed by the elastomer, for the double amplitude, at this acceleration and at the set frequency is calculated as follows:

\[ x_{3\sigma} = \frac{g_{3\sigma}}{f_n^2 \cdot 0.051} = 0.5161 \text{ inch} = 13.1 \text{ mm}. \]  
(11)

The elastomeric insulator must have sufficient elasticity (deflection), in order not to cause displacement to the limit. For this we calculate the value of the input acceleration and its displacement, at level 3 (\( g_{3\sigma} \) and \( x_{3\sigma} \)). Knowing the response to the vertical acceleration and, by referring to the resonance transmissibility, we identify the corresponding input value:

\[ g_{\text{zin} \ 3\sigma} = g_{3\sigma} : T_r = 2.96 \text{ m/s}^2. \]  
(12)

The performance of the elastomer changes with the dynamics of the vertical acceleration. Thus, the \( x_{\text{in} \ 3\sigma} \) motion is calculated for the double amplitude:

\[ x_{\text{in} \ 3\sigma} = \frac{g_{\text{zin} \ 3\sigma}}{f_n^2 \cdot 0.051} = 0.258 \text{ inch} = 6.55 \text{ mm}. \]  
(13)

Considering the characteristics of the rugged terrain, the elastomer should also be analyzed in terms of shock behavior. For the case of rough terrain, it is considered that the vertical acceleration oscillation has the form of a ramp signal. The transmissibility is described by the relation:

\[ T_{rs} = 3.2 \cdot f_n \cdot t_0 \]  
(14)

where: \( f_n \) = natural shock frequency (Hz);  
\( t_0 \) = duration of shock (s).

The rigidity of the insulator is estimated taking into account the dynamic modulus (260 psi) associated with the vibration, for the double amplitude and the static amplitude (100 psi), of the elastomer:

\[ K = \frac{100}{260} \cdot K' = \frac{100}{260} \cdot \frac{f_n^2 \cdot W}{9.8} = 7.057,1 \text{ N/m}. \]  
(15)

According to the relation (5) the shear rigidity of the elastomer is calculated:

\[ K'_{sc} = 1.4 \cdot 7.057,1 = 9.880 \text{ N/m}. \]  
(16)

Then the natural frequency of the shock is determined:

\[ f_{sc} = 3.13 \sqrt{\frac{K'_{sc}}{M}} = 11 \text{ Hz} \]  
(17)

For duration of shock \( t_0 = 0.015s \), with relations (14) and (17) is determined the value of the transmissibility of the shock (\( T_{rs} \)):

\[ T_{rs} = 3.2 \cdot 11 \cdot 0.015 = 0.528 \]  
(18)

The calculations show that the properties of the elastomer correspond to the purpose for which it was chosen. By virtual simulation, its behavior and its advantage are analyzed, for each of the three selected cases.
4. Virtual determinations regarding the behavior of the elastomer, in the suspension systems

The AMESim[5] program and the working characteristics of a LORD magnetorheological shock absorber [6] were used for the virtual modeling. Figure shows the stages of developing the virtual analysis model. Section(A) represents the classic system of passive damping. From this model, the controlled depreciation system (B) is constructed. Later, this model is associated with the Skyhook control system (C). In the analysis process, the input signal in the system is a pseudo-random type, having the specific characteristics of the road in which the data was acquired.

Figure 4. The creation of the analysis model of the suspension system

From this model, the system was developed that corresponds to the use of an electronically controlled shock absorber - magnetorheological (figure 5). In this model, different data analysis systems have been introduced regarding its operation, as well as the PID control module.

Figure 5. The electronically model for the controlled suspension analysis
The following graphs demonstrate how work the suspension system. Comparisons of the classic suspension system, of passive type, with the controlled one are made. The aim is to reduce the amplitude of the displacement of the suspended and non-suspended masses.

Figure 6 presents the behavior of the suspension system, at the level of the car body. There was recorded a substantial reduction in the amplitude of the vertical oscillation.

![Figure 6](image)

**Figure 6.** The working characteristic of the passive-active depreciation system.

In figure 7 it is observed the behavior of the suspension of the vehicle, at the seat level.

![Figure 7](image)

**Figure 7.** Variation of the vertical oscillation, at the seat level.

According to the graphs, there is an improvement in the behavior of the suspension system. Reducing the value of the vertical oscillation leads to a lower level of discomfort, so superior comfort for the driver.

For the correct observation of the efficiency of the designed damping system, the resulting vertical oscillation was analyzed, at the level of the driver of the vehicle (figure 8).
Figure 8. The amplitudes of the vertical oscillation, recorded on the driver's seat.

In order to materialize the research, the elastomer response at the frequency of 11 Hz was observed. Thus, in figure 9 we observe how the elastomer behaves, from the point of view of the damping, at the frequency resulting from calculations.

Figure 9. The damping response of the elastomer at the frequency of 11 Hz

The response of the elastomeric component is prompt and efficient in case of vehicle travel on uneven roads. This highlights its ability to decrease the level of vertical oscillation, in the established frequency range.
5. Conclusions

The design of the suspension system involves establishing the elastic and damping constants of the vehicle, in accordance with the performances they are intended to be obtained, on a certain category of road. The elastic characteristic of the suspension system influences the frequencies of the own vibration of the suspended masses of the vehicle. According to these, the stiffnesses and static arrows of the suspension springs are determined.

The optimum elastic characteristic of the suspension is obtained by the use of additional elastomeric elements or by a system of automatic adjustment of the damping of the oscillation. The efficiency of the elastomeric component is associated with the controlled shock absorber and the control strategy, for its functioning. The optimum degree of damping is obtained by adopting reduced values of the amplitudes of the vertical vibrations.

The calculations made and the results obtained demonstrate the efficiency of the solution adopted, but also its usefulness. As an additional element of the suspension system, the elastomeric component can be adopted depending on the vehicle, but also on the specificity of its destination.

References:

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