SIMULATION AND EXPERIMENTAL TESTING OF ADAPTIVE SUSPENSION DAMPING CONTROL DEPENDING ON THE FREQUENCY OF A SINUSOIDAL KINEMATIC INPUT

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Summary

A concept of adaptive suspension damping control, depending on the frequency of a sinusoidal kinematic input, has been described. This control is based on the relation between the gain function of the suspension system frequency response and the damping rate and input frequency. Due to this relation, the gain function attains its minimum values for either the strongest or the weakest damping, depending on specific input frequency bands. Appropriate changes in the damping level make it possible to obtain a curve representing the minimum gain function values that would be a combination of segments of frequency response gain curves determined for the lowest and highest damping levels available.

An idea of this concept, an algorithm of its practical implementation, and results of testing an application of this concept to a mathematical quarter-car suspension model and of using this application to control changes in the damping level of a physical quarter-car suspension model coupled with an electrohydraulic vibrator have been presented. A technical implementation of this control concept revealed considerable importance of the sub-algorithm used to identify the input frequency; a unique solution for the construction of this sub-algorithm has been proposed.

The results obtained from the tests carried out have confirmed the control concept adopted to be reasonable. The research works carried out have revealed detailed problem areas related to choosing in practice the control criterion that would influence the selection of the damping switching points.

Keywords: dynamics of automotive suspension system, variable-damping control in vehicle suspension system

1. Introduction. Suspension tasks and characteristics

To minimize the impact of road surface roughness on the generation of motor vehicle body vibrations affecting the vehicle occupants and the cargo transported, two-stage vibration isolation systems are used, consisting of pneumatic tyres and vehicle suspension system. In the suspension systems, the guiding, springing, and damping components may

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be distinguished. The vertical dynamics chiefly depends on suspension and tyre stiffness and damping characteristics. They must be appropriately selected so that desirable dynamic suspension characteristics are obtained. The latter describe, in the form of frequency-dependent functions, the way of transforming the inputs applied to the suspension system into the responses taken as a basis for evaluation of the suspension system in terms of safety and comfort. This has been illustrated in Fig. 1, where other external inputs, i.e. input forces, are also taken into account [10].

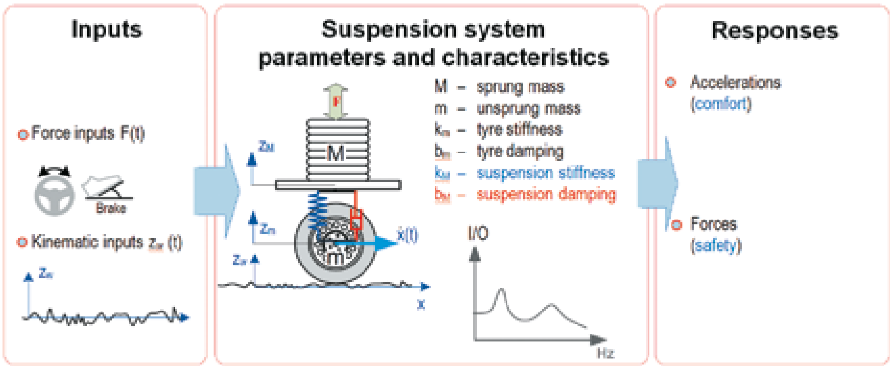

Fig. 1. Vehicle suspension as a dynamic system

The force inputs come from inertia forces, which are generated in result of changes in the longitudinal and lateral vehicle dynamics, caused by vehicle manoeuvres performed by the driver with the use of vehicle controls. Nevertheless, the inputs that most frequently occur are kinematic inputs caused by vehicle drives on uneven road surfaces with different speeds.

The vehicle responses to the kinematic inputs that normally occur during vehicle drives chiefly depend on suspension system characteristics. Among numerous responses, which are variables describing vertical vehicle dynamics, those considered most important are the quantities whose values are related to vehicle comfort or safety assessment. The comfort may be assessed by analysing the vehicle body or seat accelerations acting on vehicle occupants. To assess the safety, the vertical pressures exerted by vehicle wheels on the road surface and dynamically changing during the vehicle drive are analysed. This is connected with the role played by these forces in the possible generation of tangent forces, i.e. the adhesion forces important for driver's control over the longitudinal and lateral vehicle dynamics.

The dynamic characteristics are shaped by appropriate selection of all the parameters and characteristics of the suspension system. For practical reasons, however, the greatest possibilities are available to the vehicle designer in the field of selecting stiffness characteristics of spring elements in the suspension system and shock absorber damping characteristics. The selection of vehicle body mass and unsprung masses often depends on other vehicle design factors, e.g. total mass of other vehicle parts, vehicle body, and components of vehicle running gear, driveline, and braking system [4].

In the suspension systems commonly used in modern motor vehicles, the selection of characteristics of spring and damping components takes place at the vehicle design stage and such characteristics are assumed as constant for the whole vehicle service life (they may change due to operational wear and tear, but such changes are detected within routine inspections and, theoretically, their effects are eliminated during vehicle repairs).

The selection of appropriate system characteristics depends on such factors as:

- level of kinematic inputs, determined by the road surface type and the speed of the vehicle driven on the road with such a surface;
- level of force inputs, determined by the intensity of changes in the variables of longitudinal and lateral vehicle dynamics;
- variability in parameters of the dynamic vehicle system, especially variability in the sprung mass related to the variability in the operational vehicle load.

These factors vary during the whole vehicle service life, and within relatively wide limits at that [8, 12], and this makes it considerably more difficult to select a single optimum value for each of the stiffness and damping characteristics.

Therefore, it was as early as when friction dampers were used as shock absorbers (i.e. in 1910–1925 [2]) that a possibility of the damping force being adjusted by the driver began to be offered, with entrusting thus the driver with a task to decide about the best shock absorber setting in specific driving conditions.

Late 1980s, but practically the end of 1990s and the first decade of the 21st century, may be characterized as a period of development of shock absorbers with adjustable damping, electronically controlled [2, 5]. The presence of shock absorbers of this kind in the market is an incentive to develop shock absorber control methods that would make it possible to utilize the variability of damping settings for the vehicle ride safety and comfort to be improved. The automatic damping control methods may be divided into semi-active and adaptive [3]. The former may be used in most cases in a limited frequency range because of the time of shock absorber response to control the system operation; the latter can satisfactorily function in a wider frequency range, even if the shock absorber response time prevents the implementation of semi-active strategies.

In this paper, a concept of adaptive control has been presented, which adjusts the damping to the frequency of a kinematic input in order to improve the comfort and safety of vehicle ride.

2. The concept of controlling the damping level according to the input frequency

According to the damping control concept described herein, the dynamic characteristics of the suspension system are to be so adjusted that the frequency response gain function values would be kept as low as possible in the current input conditions. This control concept has arisen from an analysis of the gain function curves for high and low damping levels, i.e. the extreme damping levels available within the range of variability of the shock absorber damping curve. The gain function curves show that for the low damping level, the gain values are better (lower) for the input frequencies being outside of the resonance frequency range while for the high damping level, better gain values only occur when the input frequencies are within the resonance range.

This feature of the suspension system can be improved by changing the damping level according to the current input frequency. Such a concept will best work at harmonic inputs; at polyharmonic or random inputs, the effects will not be so conspicuous, unless a dominating frequency can be clearly identified. A concept of applying a control system of this kind to improve the comfort of a machine operator seat was mentioned in publication [1], but without presenting any idea of its implementation.

Additionally, it would be reasonable to take also into account the amplitude criterion, because the vehicle response to kinematic inputs depends not only on dynamic characteristics of the suspension system but also on the input proper and in the case of small input amplitude values, the use of a low damping rate might be advisable.

The general idea of the concept presented above has been illustrated in Fig. 2 for a selected tyre deflection gain function related to the safety assessment, according to the

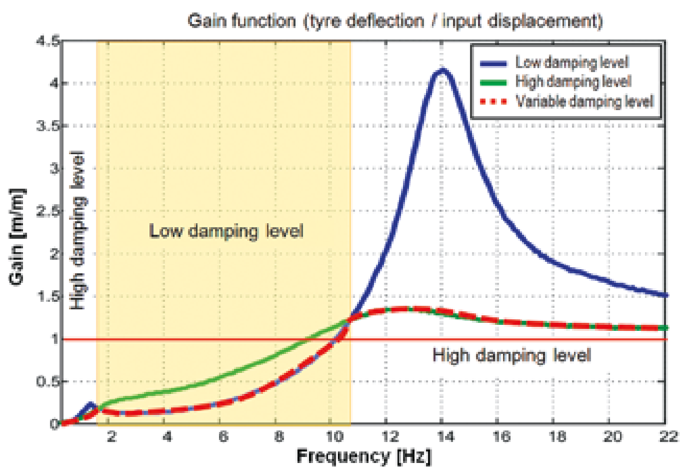

Fig. 2. The idea of changing the damping level according to the input frequency

proportionality of tyre deflections and dynamic loads in the linear range of operation of the suspension system.

It can be noticed that the gain function value at the frequencies where the damping level should be changed does not depend on the damping level. In English, such points are referred to as "invariant points" [7]. They are characterized by invariability of the gain function amplitudes regardless of the damping level. Each of the gain functions that are important for the assessment of suspension systems, i.e. that represent sprung mass accelerations, suspension deflections, and dynamic loads, has a different number of such points and the points are differently situated. The gain functions for sprung mass accelerations $H_{\ddot{z}_{\mathrm{M}}}(s)$, dynamic loads $H_{F_{\mathrm{dyn}}}(s)$, and suspension deflections $H_{(z_{\mathrm{M}}-z_{\mathrm{m}})}(s)$ have 3, 2, and 1 points of this kind, respectively [9]. Due to the location of these points at different frequency values, the optimum effect of controlling the damping characteristics cannot be simultaneously obtained for all the important indicators of the quality of operation of the suspension system.

Thanks to the invariability of the gain function values at such points, considerable changes may be made in the damping level at the input frequencies defined by the said points without causing additional phenomena related to a considerable change in the damping force as observed e.g. in the case of the two-state skyhook control strategy [14].

3. The algorithm of controlling the damping level according to the input frequency

For the control concept presented to be put into practice, an algorithm of the controlling process had to be prepared and implemented in the off-line simulation environment for simulation tests to be carried out and in the real-time simulation environment to test a prototype suspension system with adjustable damping and adaptive damping control.

The fulfilment of the following two tasks was essential for the algorithm prepared (Fig. 3):
- Detection of the predetermined input frequency;
- Making of a decision about the damping level currently needed.

As regards the second task, the making of a correct decision is facilitated by the fact that one of the two damping states, i.e. the lowest or the highest damping level, has to be selected from the characteristics of the shock absorber used. This is consistent with the previous analysis of the gain function curve. The structure of the algorithm prepared and tested has been graphically presented in Fig. 3.

The damping switching points were determined for the frequencies for which the transfer function curve does not depend on the damping rate ("invariant points").

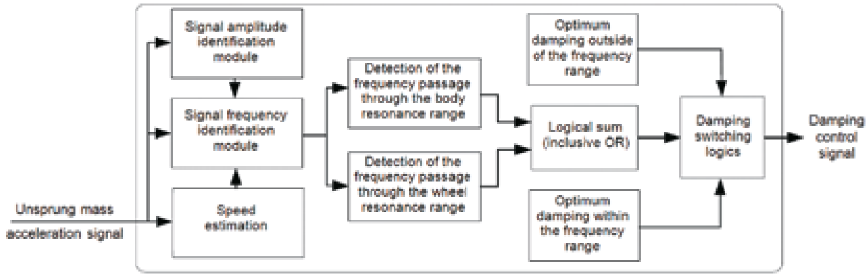

Fig. 3. Structure of the algorithm of adaptive damping control according to input frequency

3.1. Frequency identification

For the predetermined input frequency to be properly detected, the primary task was to build an algorithm that would identify the input frequencies based on signals obtainable from measurements. This limitation is inasmuch important as although the influence of input frequencies on dynamic characteristics was analysed, the input frequency cannot be directly determined because this input cannot be measured at all. Instead, only a measurement of the suspension system response can be used as a basis for estimation of the input frequency, which is identical to the output frequency in the case of forced vibration.

The response data that can be obtained by measurements are accelerations of the unsprung and sprung masses and suspension deflection. However, a problem lies in the fact that the system response amplitudes, i.e. the amplitudes of accelerations of the unsprung and sprung masses, change very much with changes in the input frequency (vehicle drive speed) and road surface type. For the unsprung mass, the acceleration amplitudes may vary by up to about 1.3 g; for the sprung mass, changes in these amplitudes may even reach 13 g. The maximum values occur only occasionally; for typical road surface types, they are far less, but the relation between vehicle body and wheel accelerations remains roughly on the same level. For this reason, the unsprung mass acceleration signal was chosen as the signal that would be valuable for frequency identification needs. This signal was taken as an input for the frequency identification algorithm. The structure of this algorithm, built to implement the idea of real-time frequency identification, has been shown in Fig. 4. The real time is defined as the algorithm calculation time that would be short enough for the information obtained from the algorithm to be usable for modification of the process for which this information was obtained.

The frequency identification method was based on an analysis of the vibration period length, with two signals, i.e. the acceleration signal for higher frequencies and the velocity signal for lower frequencies, being used for this purpose. The latter was actually a velocity estimate, obtained from the measured acceleration by pseudo-integration [6, 11]. It was used because the signal had to have distinct amplitudes so that the instant when the signal crossed the zero level could be reliably identified and a marker of this instant could be generated. At low velocities, the amplitudes of the velocity signal are more clearly visible than those of the acceleration signal.

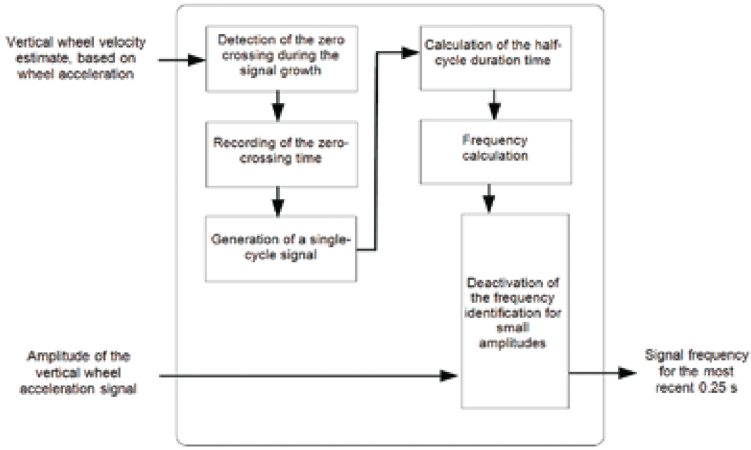

Fig. 4. The frequency identification algorithm proposed, which was verified by simulation and experimental testing

3.2. Amplitude identification

For high vertical acceleration values, the frequencies cannot be identified due to the unavailability of a signal of adequate quality, as the signal levels are very low, comparable with the background noise. For this reason, undesirable effects in the system control would be obtained from an analysis of such a signal, which would provide misleading information. Therefore, amplitude values were also analysed and the analysis results provided grounds for the frequency identification results to be invalidated for the amplitudes found insufficiently high and to be replaced with a constant value, for which low damping was adopted.

The amplitudes were identified by calculating the root-mean-square (RMS) signal value and estimating the amplitude value on these grounds (Fig. 5). The algorithm calculates the RMS value of the signal for a short analysis time adopted, equal to 0.5 s; for higher frequencies, the calculation is done on the grounds of a few periods, but for the lowest frequencies, the possibility of correct estimation of the amplitude is limited due to this.

This problem manifests itself as incorrect identification of the vibration amplitude in the initial phase of the signal curve for low frequencies (Fig. 6). However, the value of this very low frequency is of no practical importance for the damping control because the first damping switching point is situated at a frequency higher than 1 Hz.

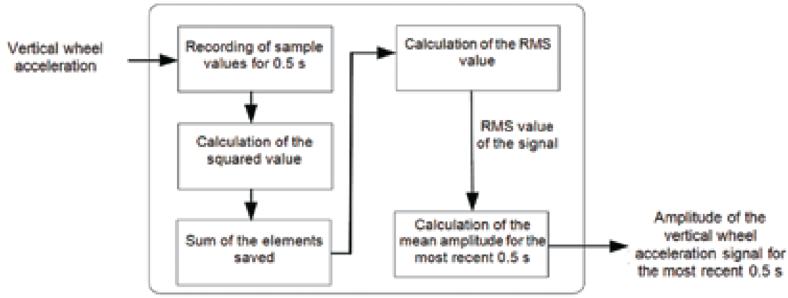

Fig. 5. Schematic diagram of the algorithm of real-time amplitude detection

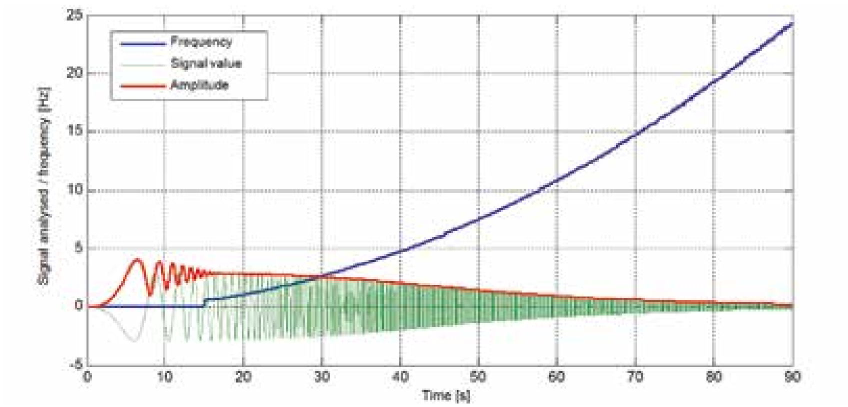

Fig. 6. Result of the identification of signal frequency and amplitude

4. Simulation testing of the algorithm of the adaptive damping control

Model tests of the possibilities of shaping the dynamic characteristics of a suspension system were carried out with the use of a method of simulation tests and frequency-domain analysis of the simulation test results on a non-linear suspension system model implemented in the Matlab/Simulink environment. The simulation tests were carried out on a model that represented a physical quarter-car suspension model. The non-linearity of the following was taken into account in the simulation model:

- stiffness characteristic curves of the suspension system, inclusive of flexible bump-stops;
- friction in the suspension system;
- damping characteristic curves;
- damping control algorithms.

The characteristic curves of the spring elements have been presented in Fig. 7 and the method of testing them and of analysing the test results has been more comprehensively discussed in publication [13]. The other model parameters were as follows: vertical tyre stiffness k_m = 290 kN/m; sprung mass M = 365 kg; and unsprung mass m = 41 kg.

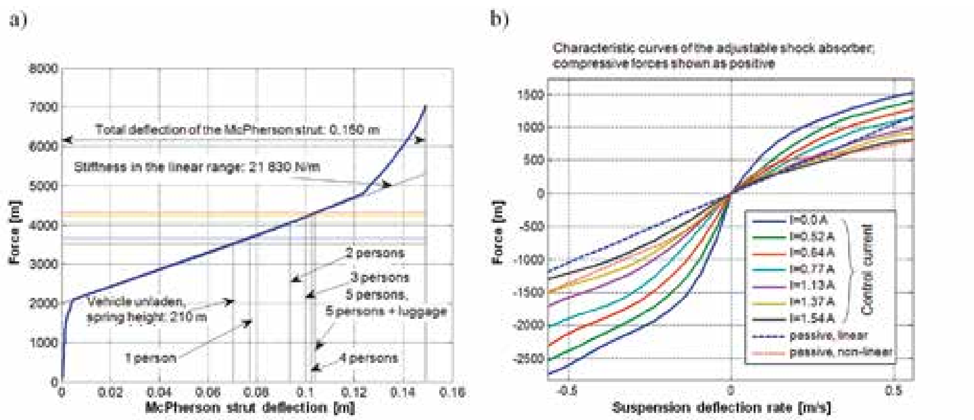

Fig. 7. Characteristic curves of components of the model of the passive quarter-car suspension system: a) spring, with loads specified; b) semi-active and passive shock absorber

The model of the suspension system was described by a pair of second-order differential equations of motion:

$$\ddot{z}_{\mathrm{M}} = {}^{1}/_{M} \cdot (F_{k_{\mathrm{M}}} + F_{c_{\mathrm{M}}} + F_t) - g \,,$$

$$\ddot{z}_{\mathrm{m}} = {}^{1}/_{m} \cdot [(F_{k_{\mathrm{m}}} + F_{c_{\mathrm{m}}}) - (F_{k_{\mathrm{M}}} + F_{c_{\mathrm{M}}}) - F_{\mathrm{t}}] - g \,, \tag{1}$$

where:

g – acceleration of gravity, g = 9.81 m/s^2;
$\ddot{z}_{\mathrm{M}}$ – vertical accelerations of the sprung mass;
$\ddot{z}_{\mathrm{m}}$ – vertical accelerations of the unsprung mass;
$F_{k_{\mathrm{M}}}, F_{k_{\mathrm{m}}}$ – elastic forces of the suspension system and tyre, respectively:

$$F_{k_{\mathrm{M}}} = k_{\mathrm{M}}(z_{\mathrm{m}} - z_{\mathrm{M}}),\ F_{k_{\mathrm{m}}} = k_{\mathrm{M}}(h - z_{\mathrm{m}}) \,, \tag{2}$$

$F_{c_{\mathrm{M}}}, F_{c_{\mathrm{m}}}$ – damping forces of the shock absorber and tyre, respectively:

$$F_{c_{\mathrm{M}}} = c_{\mathrm{M}}(\dot{z}_{\mathrm{m}} - \dot{z}_{\mathrm{M}}),\ F_{c_{\mathrm{m}}} = c_{\mathrm{m}}(\dot{w} - \dot{z}_{\mathrm{m}}) \,, \tag{3}$$

F_{t} – friction force in the suspension system, described by the Coulomb model with the static friction and the Stribeck effect being taken into account:

$$F_{\mathrm{t}} = F_{\mathrm{C}}\mathrm{sign}(\dot{z}_{\mathrm{m}} - \dot{z}_{\mathrm{M}}) + S_{\mathrm{St}}e^{-k|(\dot{z}_{\mathrm{m}} - \dot{z}_{\mathrm{M}})|}\mathrm{sign}(\dot{z}_{\mathrm{m}} - \dot{z}_{\mathrm{M}}). \tag{4}$$

with F_{C}, S_{St} and k representing the Coulomb friction force, Stribeck factor, and Stribeck velocity coefficient, respectively.

Tests of the physical model of the suspension system revealed dry friction of about 70 N to develop in the system.

The values of forces $F_{k_{\mathrm{M}}}$ and $F_{c_{\mathrm{M}}}$ were interpolated in the model from the characteristic curves presented in Fig. 7, based on the current suspension deflection $(z_{\mathrm{m}} - z_{\mathrm{M}})$ and suspension deflection rate $(\dot{z}_{\mathrm{m}} - \dot{z}_{\mathrm{M}})$.

Apart from the suspension model, the control algorithm to switch-over the damping level between the two extreme settings, i.e. the minimum (min.) and maximum (max.) damping levels available for the shock absorber under tests, was modelled as well.

A sinusoidal input with varying frequency and amplitude, presented in Fig. 6, was applied to the quarter-car suspension model. Such an input was chosen because of the technical limitations to the measurements and generation of inputs on the test rig used for the experimental research; the limitations have been more comprehensively described in publication [10].

The effect of functioning of the algorithm proposed, in the form of changes in the damping level, and the tyre deflection transmittance function have been presented in Fig. 8. These curves recorded for the suspension system adaptively controlled have been compared with corresponding curves plotted for the suspension system with the minimum and maximum damping levels having been set.

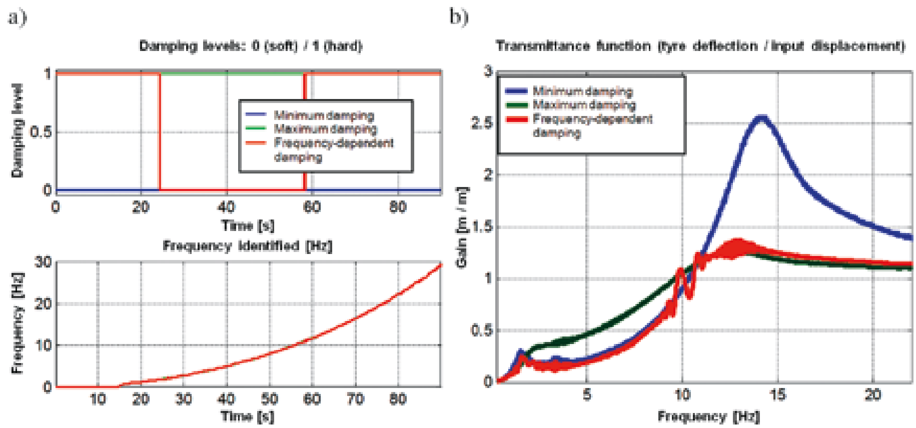

Fig. 8. Effect of switching-over the damping level according to the input frequency; comparison with the suspension performance curves plotted for the suspension system with the damping having been set to the minimum and maximum levels of the shock absorber performance range: a) changes in the damping level; b) tyre deflection gain curves

The frequency response gain curve (red) obtained in result of adapting the damping level to the kinematic input frequency clearly shows that the tyre deflection gain curve can be so shaped that the resultant curve would represent the minimum gain values obtainable in all the frequency ranges.

Due to the fact that the invariant points in the frequency response gain curves (the points where the gain value does not depend on the damping level) are situated at different frequency values for other variables considered, the gain curves obtained for the other outputs were only partly minimized. This has been shown in Fig. 9, which covers the gain functions determined for vehicle body accelerations and suspension deflections.

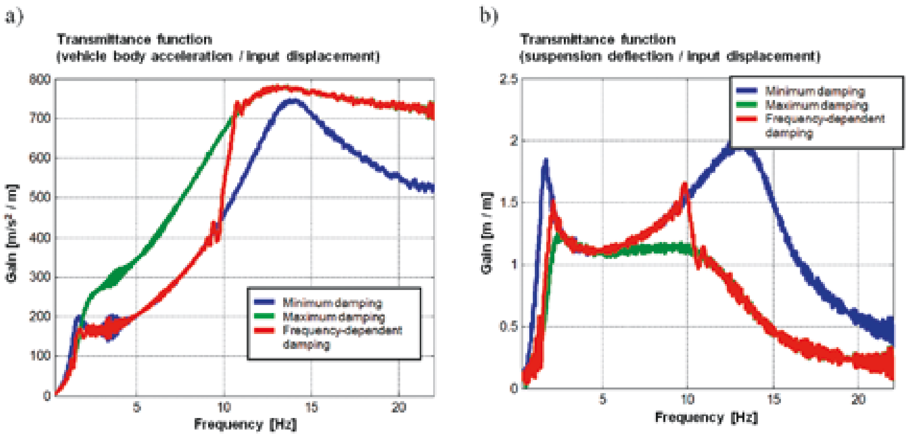

Fig. 9. Comparison of the performance curve of the suspension system adaptively controlled with the corresponding suspension performance curves plotted for the suspension system with the damping having been set to the minimum and maximum levels of the shock absorber performance range: a) for vehicle body accelerations; b) for suspension deflections

5. Rig testing of the frequency-dependent adaptive damping control strategy

The damping control algorithm presented was also experimentally verified by testing a prototype controller of a shock absorber with variable damping in cooperation with a physical quarter-car suspension model as shown in Fig. 10a). A result of the input amplitude and frequency detection for a test signal during the rig tests, with a presentation of on-line detection of the input frequency and amplitude, can be seen in Fig. 10b).

During the experimental research, smooth changing of the damping level between the maximum and minimum values was also tested, but the improvement in the ride comfort and safety was found to be less effective in this case. The lower effectiveness manifested itself in delayed drop in the gain function values for low frequencies, which can be seen in Fig. 11.

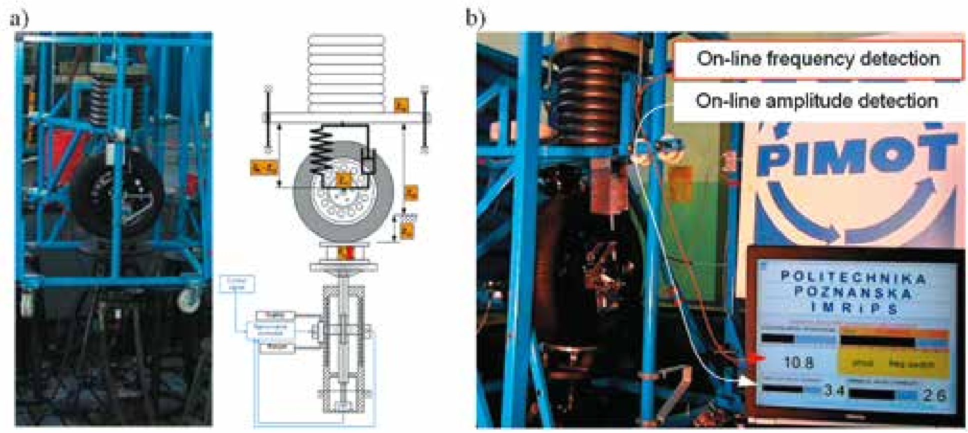

Fig. 10. Test rig for the testing of a physical quarter-car suspension model coupled with an electrohydraulic vibrator (a) and an instrument with a real-time result of the signal frequency identification algorithm being displayed on the instrument screen by the program used (b)

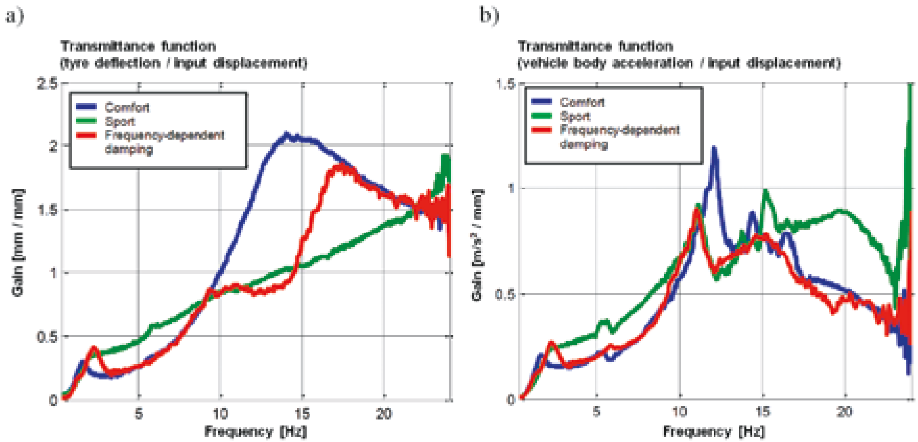

Fig. 11. Effect of smooth adjusting of the damping level to the input frequency; comparison with the suspension performance curves plotted for the suspension system with the damping having been set to the minimum (comfort) and maximum (sport) levels of the shock absorber performance range: a) tyre deflection gain curves; b) vehicle body acceleration gain curves

It should be mentioned here that some disturbances occurred at the acceleration measurements during the experimental tests, caused by lateral movements of the test rig when the input frequency passed through the unsprung mass resonance range. Due to this, a problem arose with adequate representing of the vehicle body acceleration gain curves at frequencies close to the resonance of this mass, as it can be seen in Fig. 11b). However, the effect of changing the suspension system damping to the maximum level within the unsprung mass resonance range was clearly confirmed and the output curve obtained followed the minimum values of the gain function plotted for vehicle body accelerations (in this case, the switching points were selected in accordance with the "comfort" criterion).

6. Recapitulation and conclusions

The damping control concept presented, which was verified by simulation and experimental testing, has been found to offer wide possibilities of improving the vehicle ride comfort and safety to a significant extent at periodical kinematic inputs with variable frequency. Although real road surfaces have generally a random nature, some road surface roughness wavelengths may predominate and then a change in the vehicle speed may be accompanied by a relatively slow change in the input frequency. The algorithm proposed makes it possible to monitor this frequency and to change the shock absorber damping level according to the current input frequency band. A situation of this kind was observed when the vehicle ride comfort was analysed for the vehicle being driven on a motorway. For the vehicle being driven with speeds around 110 km/h, the setting of the shock absorber to the maximum damping level resulted in the obtaining of lower acceleration values for the vehicle body resonance frequency without deterioration in the ride comfort in the other frequency bands under consideration. This was connected with a change in the input frequency, which came close to the vehicle body resonance frequency at higher vehicle drive speeds. Simultaneously, the amplitude values for the higher frequencies were of little importance for the suspension response in the form of sprung mass accelerations because of good road surface quality.

The damping control algorithm presented shows that effective identification of current input frequency is essential for the algorithm to function correctly. The algorithm proposed herein was based on detection of the instant when the velocity signal value crossed the zero level. Such a solution works well for a sinusoidal signal. In the case of polyharmonic or random signals, the use of frequency analysis with employing the fast Fourier transform is advisable.

An additional issue that is worth consideration is the solving of the problem of choosing a criterion to maximize either the comfort or the safety, which is connected with the selection of the damping switching points. This, however, is an issue concerning a controller superior to the one that governs the adaptation of the shock absorber damping to the current input frequency; such a controller may play the role of a component controller in the integrated suspension damping control system.

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