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Behavior of a semi-active suspension system versus a passive suspension system on an uneven road surface

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1. Introduction

The current suspension systems [1] can be classified in three categories: passive, semi-active and active. Oscillation sources, such as the road surface, the suspension system or the propulsion system, are located outside the automobile's interior.

By analyzing the behavior of the semi-active suspension systems on an uneven road surface in comparison with the behavior of the passive suspension systems on the same surface, we can establish how to eliminate the main discomfort sources caused by oscillations that appear inside an automobile.

In order to remove such oscillations, various control systems have been developed lately, eliminating classic suspensions which, as it is well known [2], have linear suspension characteristics. Using suspensions with adjustable characteristics allows the obtaining of different suspension coefficients, for the compression or decompression movements, and a better adjustment to the road conditions.

In order to obtain such adjustable suspensions, the viscosity of the working fluid is altered using magneto-rheological or electro-rheological liquids [3, 4]. Under the influence of the magnetic or electric field, the liquid passes from liquid state to semisolid state in a time range expressed in milliseconds. Electro-rheological fluids have a series of disadvantages such as the requirement of a high strength of the electric field and the fact that the viscosity changes within a small range, strongly influenced by temperature. For this reasons, when semi-active suspensions are built, magneto-rheological liquids are preferred [5]. When the liquids are exposed to various magnetic fields, the viscosity changes for the magneto-rheological liquids obtained through the colloidal dispersion of fine solid metal particles in carbohydrate-based synthetic oil.

The switch frequencies of the semi-active systems are higher than those of the oscillations specific to the automobile's wheels and chassis. The semi-active systems can quickly and dynamically switch from one specific curve to another at any given point of a specific curve.

2. Suspension system model

A quality suspension must provide a good vehicle behavior and a certain degree of comfort depending on the vehicle's interaction with the road unevenness [1, 6]. When the vehicle is subjected to road elevations, the vehicle should not undergo through a great amount of oscillations and, if such oscillations appear, they must be removed as soon as possible. Designing a vehicle's suspension is a problem that requires a series of calculations depending on the intended purpose.

Until now, a series of models have been developed [3, 7-10], such as the quarter car, half car or full car suspension. Next, we will make reference to the quarter car suspension model applied to a passive suspension system (Fig. 1, a) or semi-active suspension system (Fig. 1, b), using a PID controller.

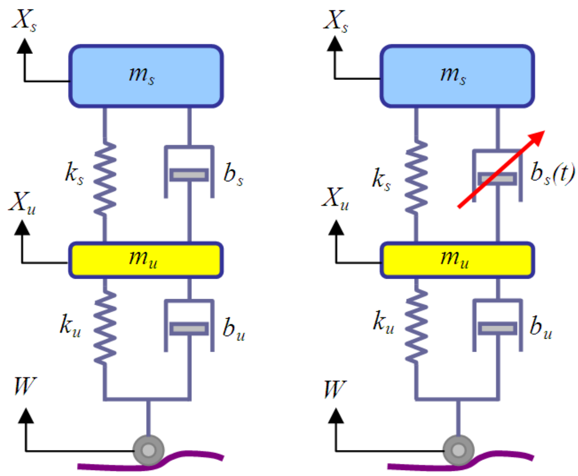

Fig. 1 Suspension system: a – passive, b – semi-active

For the systems in Fig. 1 we have sprung mass m_s, unsprung mass m_u, coefficient of elasticity for the suspension spring k_s, coefficient of elasticity for the wheel k_u, damping coefficient for the damper b_s, coefficient as a time function for the magneto-rheological damper of the semi-active suspension $b_s(t)$, damping coefficient of wheel b_u, disturbance created by the road acting upon the suspension w. Different than the passive system and semi-active system has a variable damping coefficient $b_s(t)$ which will controlled by a controller, $b_s(t)$ will be the output of the PID controller. The controlled variable will be a time function.

It is necessary to implement a gradient algorithm for adjusting the current input to the magneto-rheological damper that minimizes the square of the vertical acceleration. For the control block diagram of the semi-active suspension system, we will use as output measure the distance $x_s - x_u$. The motion equations can be obtained using Newton's second law of motion, for each of the two moving masses, and Newton's third law of motion for the masses' interaction.

$$\left.\begin{aligned} m_s\ddot{x}_s &= -b_s\left(\dot{x}_s-\dot{x}_u\right)-k_s\left(x_s-x_u\right); \\ m_u\ddot{x}_u &= b_s\left(\dot{x}_s-\dot{x}_u\right)+k_s\left(x_s-x_u\right)+ \\ &\quad +b_u\left(\dot{w}-\dot{x}_u\right)+k_u\left(w-x_u\right). \end{aligned}\right\} \quad (1)$$

The second degree differential equations for a passive suspension system have been written. Solving this system of equations is difficult as it implies using the Laplace transform. In such case, a translation from the time domain to the operational domain is necessary. The equation system becomes:

$$\left.\begin{aligned} m_s X_s s^2 &= -b_s\left(X_s - X_u\right)s - k_s\left(X_s - X_u\right); \\ m_u X_u s^2 &= b_s\left(X_u - X_s\right)s + k_s\left(X_s - X_u\right) + \\ &\quad + b_u\left(W - X_u\right)s + k_u\left(W - X_u\right). \end{aligned}\right\} \quad (2)$$

From the first equation of system (2) we obtain:

$$X_u(s) = X_s(s)\frac{m_s s^2 + b_s s + k_s}{b_s s + k_s}. \quad (3)$$

If we add up the two equations of system (2) and replace $X_u(s)$, we obtain:

$$\left.\begin{aligned} & m_s s^2 X_s(s) + X_s(s)\frac{C_1 C_2}{b_s s + k_s} = W(s)\left(b_u s + k_u\right); \\ & C_1 = \left(m_s s^2 + b_s s + k_s\right); \\ & C_2 = \left(m_u s^2 + b_u s + k_u\right). \end{aligned}\right\} \quad (4)$$

Next we define three transfer functions of interest:

$$\left.\begin{aligned} & H_1(s) = \frac{X_s(s)}{W(s)},\ H_2(s) = \frac{X_u(s)}{W(s)}, \\ & H_3(s) = \frac{X_s(s) - X_u(s)}{W(s)}. \end{aligned}\right\} \quad (5)$$

Using relations 4 and 5 we can determine, by calculus, the transfer function $H_1(s)$, obtaining:

$$\left.\begin{aligned} & H_1(s) = \frac{\left(b_u s + k_u\right)\left(b_s s + k_s\right)}{m_s s^2 B_1 + B_2 B_3}; \\ & B_1 = \left(b_s s + k_s\right); \\ & B_2 = \left(m_s s^2 + b_s s + k_s\right); \\ & B_3 = \left(m_u s^2 + b_u s + k_u\right). \end{aligned}\right\} \quad (6)$$

From the (2) equations we have the solutions of $H_1(s)$&$H_2(s)$ in Laplace Domain in terms of $\boldsymbol{S}$:

$$\left.\begin{aligned} & H_1(s) = \frac{b_s b_u s^2 + \left(k_s b_u + b_s k_u\right)s + k_s k_u}{m_s m_u s^4 + A_1 s^3 + A_2 s^2 + A_3 s + k_s k_u}; \\ & H_2(s) = \frac{m_s b_u s^3 + A_4 s^2 + \left(k_s b_u + b_s k_u\right)s + k_s k_u}{m_s m_u s^4 + A_1 s^3 + A_2 s^2 + A_3 s + k_s k_u}; \\ & A_1 = \left(m_s b_s + m_s b_u + m_u b_s\right); \\ & A_2 = \left(m_s k_s + m_s k_u + m_u k_s + b_s b_u\right); \\ & A_3 = \left(b_s k_u + k_s b_u\right); \\ & A_4 = \left(m_s k_u + b_s b_u\right). \end{aligned}\right\} \quad (7)$$

Another method that can be used for the passive suspension system (1), is based on the general form of the state-space model:

$$\left.\begin{aligned} & \dot{X} = A(t)X(t) + B(t)U(t); \\ & Y(t) = C(t)X(t) + D(t)U(t). \end{aligned}\right\} \quad (8)$$

Essential for an orderly formulated model is the choice of the states. We can choose the state variables for the displacement and velocity of sprung and unsprung mass: $\dot{x}_1 = \dot{X}_s$ absolute velocity of the sprung mass, $\dot{x}_2 = \dot{X}_u$ absolute velocity of unsprung mass, $x_3 = X_s - X_u$ suspension deflection (rattle space), $x_4 = X_u - W$ tire deflection.

In the Eq. (8) X, Y, U, A, B, C and D are the matrices of various order. A is a state space matrix, B is an input matrix, C is an output matrix, D is a direct transmission matrix and U is an input of the system.
The mentioned matrixes are:

$$A(t) = \begin{bmatrix} -\frac{b_s}{m_s} & \frac{b_s}{m_s} & -\frac{k_s}{m_s} & \frac{k_s}{m_s} \\ \frac{b_s}{m_u} & -\frac{b_s + b_u}{m_u} & \frac{k_s}{m_u} & -\frac{k_s + k_u}{m_u} \\ 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix}; \quad (9)$$

$$X(t) = \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ x_1 \\ x_2 \end{bmatrix}, B(t) = \begin{bmatrix} 0 & 0 \\ \frac{b_u}{m_u} & \frac{k_u}{m_u} \\ 0 & 0 \\ 0 & 0 \end{bmatrix}, U(t) = \begin{bmatrix} \dot{W} \\ W \end{bmatrix}; \quad (10)$$

$$\left.\begin{aligned} & Y(t) = \begin{bmatrix} x_1 \\ x_2 \\ x_1 - x_2 \end{bmatrix}; C(t) = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & 1 & -1 \end{bmatrix}; \\ & D(t) = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \end{bmatrix}. \end{aligned}\right\} \quad (11)$$

For semi-active suspensions we replace b_s with a time function $b_s(t)$ so that Eq. (1) become:

$$\left.\begin{aligned} m_s \ddot{x}_s &= -b_s(t)\left(\dot{x}_s - \dot{x}_u\right) - k_s\left(x_s - x_u\right); \\ m_u \ddot{x}_u &= b_s(t)\left(\dot{x}_s - \dot{x}_u\right) + k_s\left(x_s - x_u\right) + \\ &\quad + b_u\left(\dot{w} - \dot{x}_u\right) + k_u\left(w - x_u\right). \end{aligned}\right\} \quad (12)$$

Different than the system (1), system (12) has a variable damper coefficient $b_1(t)$ which will controlled by a controller. Solving the equation systems is extremely difficult. In order to study the suspension system behavior we often resort to a simulation created using Matlab Simulink.

3. Implementation using Matlab Simulink

In Fig. 2 using Matlab [11], we display a simulation scheme for the quarter car model, passive and semi-active suspension systems. It can be observed that there are multiple blocks whose purpose will be described from left to right.

The first block, entitled Signal Builder, has the purpose of inserting different stimulus signals that will simulate the nonconformity of the rolling surface. The next three blocks, named Controllers, are designed for the control system of the semi-active suspension. We can observe PID type controllers - proportional integrative derivative, PI - proportional integrative and PD – proportional derivative.

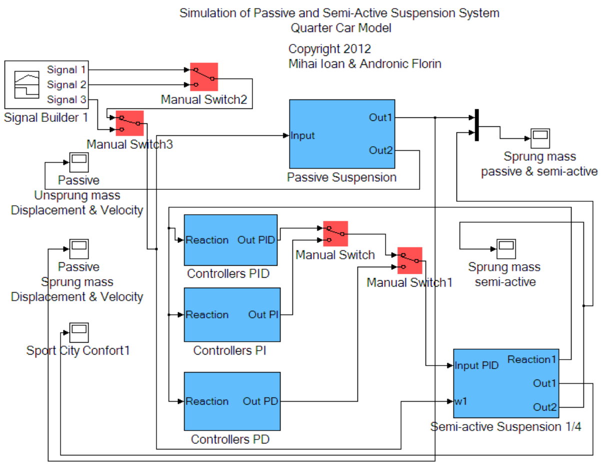

Fig. 2 The Matlab suspension simulation scheme, quarter car model

In order to use any signals or controllers in the scheme we inserted manual switches. The stimulus signal can be adjusted depending on the end purpose. We considered appropriate to study the behavior of the suspension system for a step signal, as it can be seen in Fig. 3, first case. In this way, we can observe if the simulation is carried out properly. The stimulus signal 2 corresponds to a rough road with thresholds and holes.

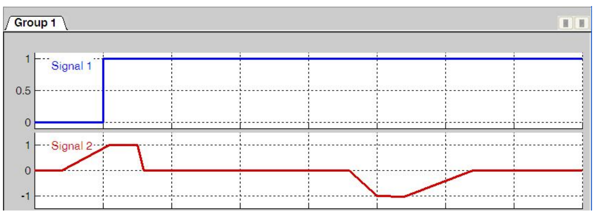

Fig. 3 Stimulus signals used for the simulation

In order to analyze how the quarter car model suspension systems behave we used as input data the following parameters, corresponding to the above mentioned equations: $m_s = 487.5$; $m_u = 58.7$; $k_s = 6000$; $k_u = 140000$; $b_s = 300$; $b_u = 1500$; $k_i = 5.52$; $k_d = 10.0$; $k_p = 0.552$. Also, in order to verify if the simulation in Matlab Simulink is carried out correctly we used two other methods of simulation, the first using as transfer function the Eq. (7) and the second, using as state space system the Eq. (8). In the passive and semi-active suspension blocks, as per Fig. 4, we inserted coefficients using matrixes for both cases, where indices 1 and 2 correspond to the sprung and unsprung mass, respectively.

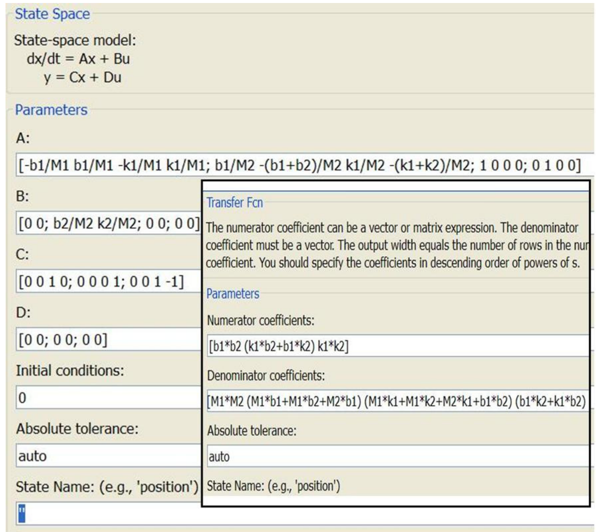

Fig. 4 Parameters for state space model and transfer function

The simulations were carried out for three situations: the scheme designed in Simulink, the state space model and the transfer function.

4. Simulation result and discussion

For passive suspension, using a step-type stimulus signal we obtained, by simulation, the plots from Fig. 5, where one can observe the sprung mass displacement and sprung mass velocity measures.

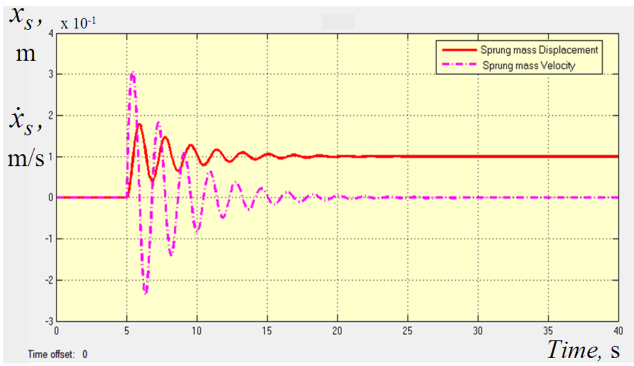

Fig. 5 Sprung mass displacement and velocity signals, passive model

We observe that when the stimulus signal is applied the sprung mass displacement measure displays typical behavior for 27 seconds, after which the oscillation amplitude is null. From a passenger comfort point of view, the severe amplitude variations that are transferred to the passengers are maintained for at least 10 seconds, noting that the first oscillations are the strongest.

Using the same parameters and the step-type stimulus signal, we obtained Fig. 6 where the unsprung mass displacement and unsprung mass velocity can be observed.

For the unstrung mass referred to in Fig. 6 we display the stimulation time, which falls between 4.75 seconds and 5.35 seconds, so that only within this timeframe significant changes to the tracked parameters occur. We observe that the unsprung mass movement displayed as a detail image on the right side of Fig. 6 shows time oscillations of 19 seconds and that the shape of the signal obtained by simulation is similar to the theoretical shape of the signal. Unlike in the situation of the sprung mass, we observe that the unsprung mass movement is approximately ten times greater in magnitude. The oscillation velocity of the unsprung mass is strongly changed when applying a stimulus signal from 5 seconds to 5.3 seconds, a timeframe which is greatly reduced. The oscillation velocity of the unsprung mass continues to display small variations until 19.7 seconds.

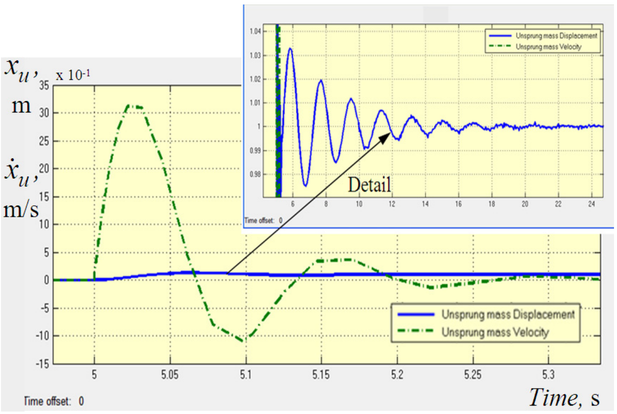

Fig. 6 Unsprung mass displacement and velocity signals, passive model

For the semi-active suspensions system we considered that such system is controlled by a PID controller - proportional integrative derivative. In order to compare results we used the same step-type stimulus and the same simulation duration. The obtained signals for the semi-active, quarter car model are displayed in Fig. 7 with a separate detail, on the right, for sprung mass displacement.

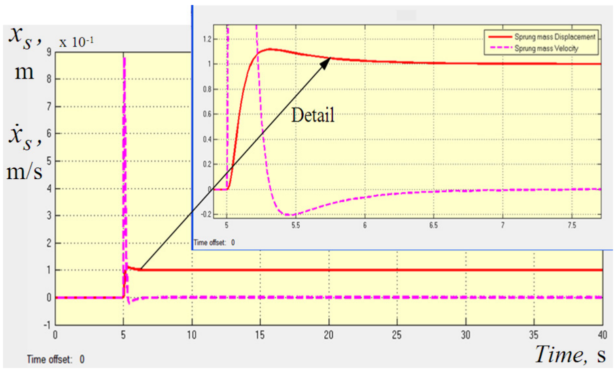

Fig. 7 Sprung mass displacement and velocity signals, semi-active model

Comparing the signals obtained by Matlab simulation for the semi-active model sprung mass (Fig. 7) to the signals obtained by Matlab simulation for the passive model (Fig. 5), we observe extremely big differences between the signals. Consequently, for the semi-active system, the sprung mass displacement is modified when the stimulus signal is applied from 5 seconds to 6.6 seconds. In such case, using the PID controller and the magneto-rheological dampers the system is stabilized in only 1.6 seconds compared to 27 seconds obtained for the passive model. Moreover, the sprung mass velocity is lower in the case of the semi-active suspension.

The simulation of the unsprung mass behavior displayed in Fig. 8 presents, for the semi-active model, how the analyzed parameters vary.

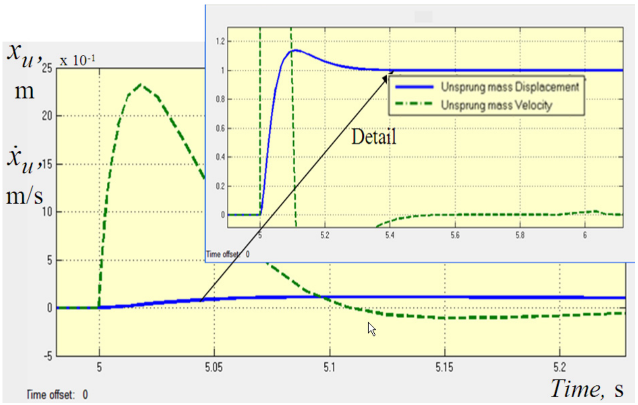

Fig. 8 Unsprung mass displacement and velocity signals, semi-active model

We observe, in this case as well, that a faster stabilization of the system occurs, compared to the passive model, including the elimination of the system oscillations which, as it is well known, affects the passengers' comfort. The unsprung mass displacement size is alleviated in only 1 second, starting when the stimulus signal was applied, whereas the spring velocity is quickly stabilized in 1.125 seconds.

The simulation using the second stimulus signal corresponds to a rolling surface with one threshold and one hole. The obtained post simulation results are displayed in Fig. 9 for sprung mass, passive model.

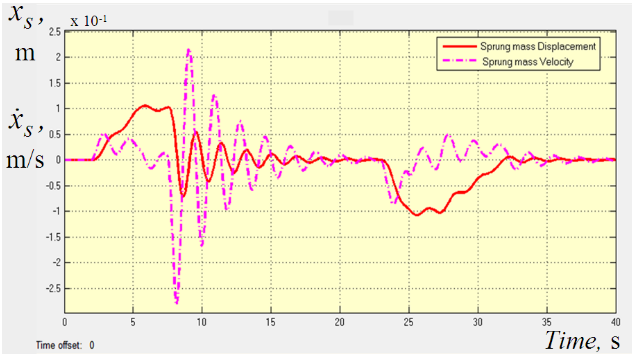

Fig. 9 Sprung mass displacement and velocity signals, case 2, passive model

By analyzing the signals that resulted from the simulation we observe that the disturbances applied to the sprung mass persist in the passive case for the entire period of 40 seconds, a fact which demonstrates that the passengers will have discomfort issues. In such case, the suspension system only partially reduces the unevenness of the road, if such unevenness frequency is high. Although a first segment of the signal was regarded as ramp signal, the period of time from when the disturbance signal appeared until the pulsations were alleviated is greater than 20 seconds, as in the case of the step-type signal. We note that, unlike the step-type signal, in the case of ramp signal, the amplitude of the sprung mass velocity magnitude is reduced from 5.6 m/s to 5.1 m/s, a fact that is explained based on the signal shape. Obviously, the step-type signal is more harmful to the sprung mass elements.

Simulating how the unsprung mass works in a passive suspension system (Fig. 10) clearly shows that modifying the unsprung mass displacement closely follows the shape of the stimulus signal in case of rolling surface unevenness. At the same time, the unsprung mass velocity

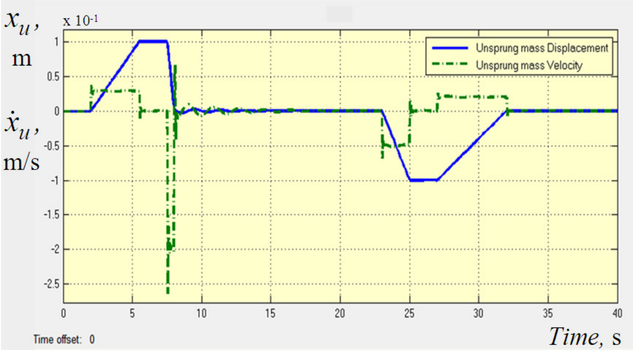

Fig. 10 Unsprung mass displacement and velocity signals, case 2, passive model

shows small variations, step by step, for the ramp areas compared to the step signal. Large increases of the unsprung mass velocity can be observed, only for large declivities, followed by straight areas of the rolling surface. As a result, when thresholds or holes exist on the rolling surface, both unsprung mass and sprung mass are strained.

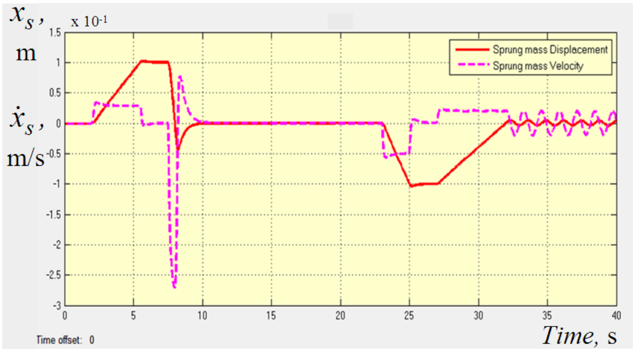

Fig. 11 Sprung mass displacement and sprung mass velocity signals, case 2, semi-active model

In Fig. 11 we display the results of the quarter car simulation for the sprung mass of the semi-active suspension system.

We immediately notice the fact that, unlike in the case of the passive model (Fig. 9), there are no more permanent oscillations for the sprung mass displacement, a fact which clearly indicates an increase in the passengers' comfort. However, it can be observed that the system follows the configuration of the road, a fact that can be avoided only by using semi-active suspension systems. At the same time, we also notice that the sprung mass' stabilization time is significantly reduced, a fact that allows immediate reestablishment of the front suspension stability, compared to the passive model.

The results of the simulation displayed in Fig. 12 correspond to the unsprung mass displacement and unsprung mass velocity semi-active model. We notice that no significant changes occur when compared with the passive model. When applying the stimulus signal and descending on the steep slope, the system alleviation is faster and no further oscillations appear, as in the case of the passive model. Although for the semi-active model the unsprung mass velocity is slightly smaller in value when descending the excitation ramp signal, for the rest of the domain there are no significant changes to the passive model. Still, one can observe that for the passive model there are a series of peaks reached by the unsprung mass velocity for each change in the stimulus signal. The size of these peaks is, however, insignificant.

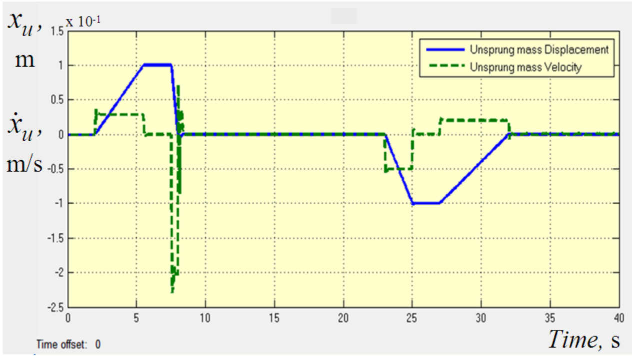

Fig. 12 Unsprung mass displacement and velocity signals, case 2, semi-active model

5. Conclusions

1. Using semi-active systems instead of passive systems leads to an almost total elimination of the system oscillations, a reduction in the amplitude of the oscillatory phenomena and a reduction in the disruptive time, which constitutes great advantage.

2. Out of simulations it resulted that the shape of excitation signal is extremely important. The more the elevations deviate from the step signal towards the ramp or slope excitation signal, the more the suspension system is less strained.

3. The damper plays a significant role in stabilizing the semi-active system and it can be controlled by modifying the applied strain. Such strain varies according to the road configuration, being obtained based on the information gathered from the vehicle's sensors. It was established that the springs have an almost similar behavior in both passive and active systems. This can be explained based on the impossibility to interfere with their characteristics. The small signal differences obtained when simulating the springs of the semi-active systems can be explained based on the disturbances that appear in the case of magneto-rheological dampers.

4. In case of big elevations in the road that take the shape of thresholds or holes, the semi-active systems drastically diminish the suspension oscillations. However, we also noticed that the semi-active systems follow the road configuration.

5. The semi-active suspension systems can be effectively employed to any type of vehicle with improved both ride comfort and steering stability.

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I. Mihai, F. Andronic

PUSIAU AKTYVIOS PAKABOS FUNKCIONAVIMO NELYGIAME KELIO RUOŽE LYGINIMAS SU PASYVIĄJA PAKABOS SISTEMA

R e z i u m ė

Šiame straipsnyje pusiau aktyvi pakabos sistema lyginama su pasyviąja, kai sužadinimo šaltinis yra važiuojamasis paviršius. Tam buvo būtina sudaryti lygtis, išreiškiančias tiek pasyviąją, tiek aktyviąją pakabų sistemą. Kiekvienu atveju skaičiavimo schemos buvo sukurtos naudojant *Matlab Simulink* programas. Modeliavimui buvo pasirinktas automobilio ketvirčio modelis. Modelio teisingumui patikrinti buvo panaudoti būsenų erdvėje ir perdavimo funkcijos modeliai. Kadangi pasyvieji slopintuvai turi tiesinę slopinimo charakteristiką, buvo tiriama pusiau aktyvi sistema, naudojanti magnetoreologinius slopintuvus su reguliuojamomis charakteristikomis. Modeliavimo diagramoje buvo panaudotas PID proporcinis, integruojantysis, diferencijuojantysis valdiklis. Naudojant dviejų tipų trikdžio signalus buvo ištirta, kaip veikia pusiau aktyvi pakabos sistema su skirtingais slopinimo koeficientais. Modeliuojant buvo nustatyti spyruoklinio slopintuvo poslinkio ir greičio atsako signalai. Gautų rezultatų analizė rodo pusiau aktyvių sistemų pranašumus, palyginti su pasyviosiomis. Tokių sistemų apribojimai lyginami su aktyviųjų sistemų apribojimais.

I. Mihai, F. Andronic

BEHAVIOR OF A SEMI-ACTIVE SUSPENSION SYSTEM VERSUS A PASSIVE SUSPENSION SYSTEM ON AN UNEVEN ROAD SURFACE

S u m m a r y

The present paper intends to study the behavior of semi-active suspension systems in comparison with passive suspension systems when subjected to oscillation sources caused by the rolling surface. For this, it was necessary to establish the equations specific to the passive and semi-active suspension systems. For each case, the simulation schemes were developed using Matlab Simulink. For the simulation, the quarter car model was chosen. In order to verify the correctness of the model, the State Space and Transfer function models were also used. As the passive dampers have a linear dampening characteristic, we studied the semi-active systems that use magneto-rheological dampers with adjustable characteristics. In the simulation diagram we used a PID controller – proportional integrative derivative. Using two types of stimulus signals we analyzed the behavior of the semi-active suspension system with different dampening coefficients. By simulation, response signals for damper/spring displacement and damper/spring velocity were obtained. The analysis of the obtained results shows the advantages of semi-active systems in comparison with passive systems and the limitations of such systems compared to active systems.

Keywords: semi-active suspension, magneto-rheological damper, PID controller, quarter car model.

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