VIBRATION INVESTIGATION OF VEHICLE EQUIPPED WITH CONTROLLED PIEZOELECTRIC DAMPERS

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Abstract

In this paper an effect of controlling damping force in a vehicle suspension system is discussed. A vehicle suspension system is equipped with controlled piezoelectric dampers (PZD) which allow controlling vehicle body vibration.

The main goal of performed and presented in the paper studies was to determine the properties of piezoelectric damper and propose control strategy. As piezoelectric damper was used hydraulic cylinder with piezoelectric valve controlling flows between chambers of the damper. Some results of experimental studies of the damper dissipation characteristics were obtained and are presented.

Vertical vibration reduction of vehicle body has an influence on comfort and safety by decreasing a value in wheel contact forces variation. Control of friction in a vehicle suspension system can be realized by controlling piezoelectric valves and flows between chambers of the PZD dampers. In this paper also some algorithms of vibration control are proposed and discussed. The main goal of this paper is to investigate issues associated with control algorithms for an adaptive (semi-active) suspension system of a vehicle equipped with new type of semi-active dampers. The paper presents results of numerical simulations of a vehicle suspension system equipped with a piezoelectric damper.

Keywords: *transport, piezoelectric damper, simulation, control algorithms, adaptive suspension*

1. Introduction

Over the last two decades, developments in vehicle suspension design have focused on increased level of safety and comfort. For years, many automotive companies have conducted research into continuous control of a vehicle suspension. These efforts have enabled rapid development of electronics and drive control systems, as well as, devices which can be employed to achieve the goal.

At the moment, thanks to advances in measurement technology and microprocessor control, the development of a new generation of smart materials may lead to wide implementation of concepts of semi-active damping systems into everyday life - especially using technology based on the properties of so called 'intelligent' materials with includes magneto-rheological, electro-rheological and piezoelectric materials (and many others).

In case of vehicles, designers have long been exploring ways to utilize so-called smart suspension systems in order to reduce vibrations. Oftentimes, the crucial part of a solution involved active or semi-active devices designed to dissipate the kinetic energy in a controlled way. Frequently, however, intelligent suspension systems have suffered from the lack of a time-control loop putting accuracy of their control in question. It has been suggested recently [1-6] that the time-control loop can be introduced into intelligent suspension systems by means of semi-active MRDs (magneto-rheological devices). Semi-active MRDs rely on the use of magneto-rheological fluids (MRF), which can significantly change their physical properties upon application of magnetic field. In late 1990s Carlson [1] developed an MRD applicable to on-and-off-highway vehicle suspension system. It was experimentally demonstrated that the capability to control the damping force in a vehicle suspension system can be easily achieved by MRDs. Many other researches [2], [5] and [6] provided additional know-how in the field of constructing effective vehicle and aircraft suspension systems with use of MRDs under different optimization criteria. At the same time, others conducted a series of works devoted to controlling the active suspension vehicle with other type of active or semi-active devices [3]. It should also be noted that the controlled hydraulic dampers are now in-use in several manufactured vehicle suspension systems, where their dissipative features are adapted in time to the current vehicle load and road conditions.

The main goal of work presented in this paper was to investigate issues associated with control algorithms for an adaptive (semi-active) suspension system of a vehicle equipped with new type of semi-active dampers. We present results of numerical simulations of a vehicle suspension system equipped with a piezoelectric damper (PZD).

2. Piezoelectric damper design

A schematic diagram of the piezoelectric vibration damper (PZD) with piezoelectric valve is shown in Fig. 1, where the components of device are also presented. The PZD is built as a hydraulic cylinder with double side acting piston and single rod. Piezoelectric valve is built in as a hydraulic cylinder bypass. As piezoelectric stack was used piezoelectric actuators PPA80L from Cedrat Technologies (Cedrat Group).

PZD damper chambers (cf. Fig. 1) are filled up with hydraulic oil. The chambers are sealed off from each by the piston (3). When the piston moves the liquid flows through the hydraulic hose and the piezoelectric valve (1). The flow of liquid through the controlled gap (5) in piezoelectric valve is accompanied by energy dissipation. The gap in the piezoelectric valve can be controlled by voltage applied to the piezoelectric stack (7). Subsequently, these changes of the voltage in the piezoelectric stack can alter the internal friction of liquid flowing through the controlled gap, leading to a change in the dissipation characteristic of the PZD.

The most important feature of the PZD damper is the possibility of quick and easy forming dissipative adaptive characteristics, allowing efficient and controlled energy dissipation process in mechanical structures. The PZD damper can quickly change damping forces due to short lag of a mechanical system response to electric signal. These features allow using the PZD damper in mechanical systems, where quick change is needed. The problem of controlling proprieties of PZD damper in mechanical systems is simplified to the selection of damping forces in the PZD damper by adjusting electric voltage signal applied to piezoelectric stack.

3. Experimental investigation of PZD damper

The goal of the first part of our experimental studies was to determine the dissipation characteristics of the PZD damper treated as a whole device (black-box). Those characteristics can be used to identify parameters of device rheological model.

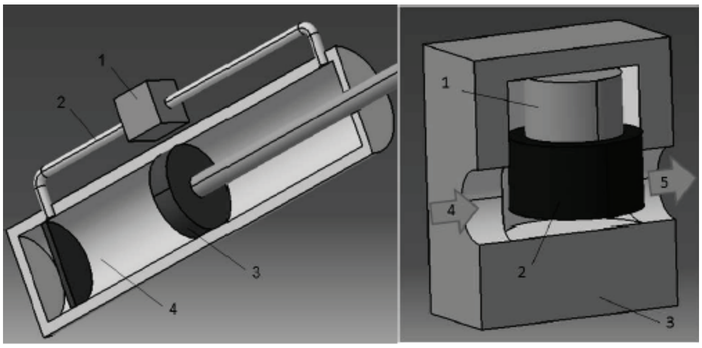

Fig. 1. Simplified construction diagram of piezoelectric damper and piezoelectric valve

Fig. 2. General view of employed and future experimental setup

The experimental studies were carried out on a dedicated stand with kinematic excitation induced by a hydraulic actuator. The damper under investigation was mounted into the experimental stand and displacement and velocity measurements were taken by recourse to computer data acquisition system. Fig. 2 shows a general view of the testing stands used for rheological model parameter identification and prepared experimental stand which will be used to study efficiency and effectiveness over PZD devices control algorithms.

The first stage of experimental investigation was successful and dissipation characteristics of PZD damper were achieved. The dissipative characteristics of the PZD device are plotted in Fig. 3 as graph of displacement vs. force and as velocity vs. strength. The displacement vs. force graph clearly displays the venerable hysteresis loop, while the velocity vs. strength graph directly represents the device dumping characteristics achieved for different control signal applied to controller responsible for controlling gap in piezoelectric valve. These results were obtained at sinusoidal forcing with 1Hz frequency and amplitude approximately of 20-23 mm. Those results can be used to identify rheological model of PZD damper.

4. Vehicle model

Mathematical model of the vehicle used to investigate control algorithm in the form of a rigid plate is shown in Fig. 4. This simplified vehicle model is described by oscillating mass m and mass moments of inertia Iy, position of centre of mass a1 and a2. The parameter xk specifies the location of the checkpoint K (for example driver position).

The suspension system of vehicle is also described by spring stiffness - k1 and k2, and shock absorbers with damping coefficients - c1 and c2. Vibrations of the vehicle model are caused, as during experimental studies described in previous part of article, by kinematics extortion described by functions ξ(t)1 and ξ(t)2.

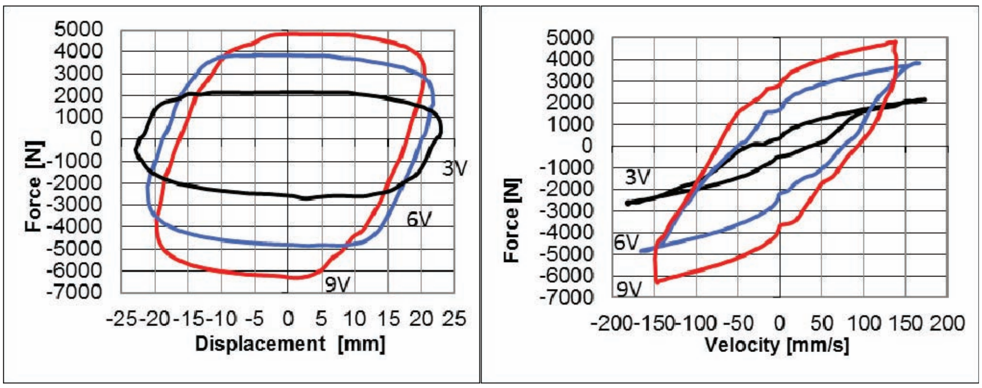

Fig. 3. The hysteresis loop and characteristics of the damper - results of measurements of the PZD damper generated with different control signals and at frequency of 1 Hz and amplitude of 20-23 mm

The vehicle model has been described in the coordinates:

$$X = \left[z, \Phi_y\right]^T, \tag{1}$$

where:
X1 = z - vertical displacement of vehicle,
X2 = Φy - angle of rotation around the transverse axis of the vehicle.

Flat model of the vehicle has been described by means of the following set of ordinary differential equations:

$$M\ddot{X} + H(S+T) = 0, \tag{2}$$

where M = diag(m, Jx,Jy).

Spring force S and friction force T in damper are described by means of the following relationship:

$$S = [S_1, S_2]^T, \ \ S_i = f_i(U_i), \tag{3}$$

$$T = [T_1, T_2]^T, \ \ T_i = F_i(V_i, w_i). \tag{4}$$

The values of suspension displacements U_i and velocity V_i (i=1,2) are calculated from following formulas:

$$U_i(t) = H_i^T \cdot X_i(t) + \xi_i(t), \qquad 5)$$

$$V_i(t) = H_i^T \cdot \dot{X}_i(t) + \dot{\xi}_i(t), \tag{6}$$

where ξi (i=1,2) are functions describing roughness of the pavement.

Vectors H_i (i=1,2) describes the allowed configuration space to the forces S_i and T_i :

$$H_1 = \left[1, -a_1\right]^T, \quad H_2 = \left[1, a_2\right]^T. \tag{7}$$

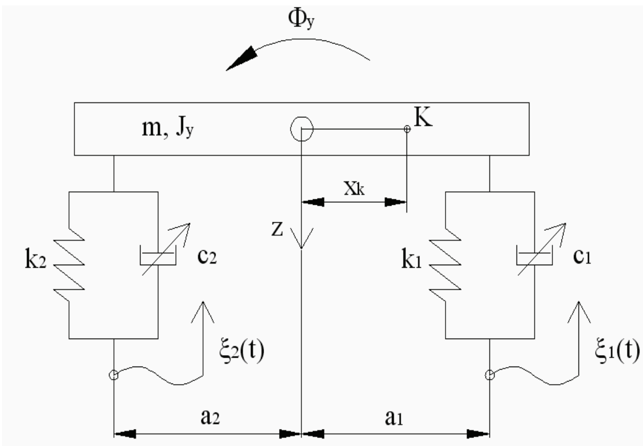

Fig. 4. Simplified model of a vehicle

The damper forces T are determined on the basis of assumed control algorithm F, which is described later. Then, the equation of vibration of the vehicle turns into the following form:

$$M\ddot{X} + HS(U) + HT = 0, \tag{8}$$

$$T(t) = F(U(t), V(t)), \tag{9}$$

where F is a function describing the algorithm for designating signals applied to piezoelectric stack.

5. Control algorithm

The control algorithm was developed, assuming the need to meet two criteria: improvement of driving comfort and reduction of wheel force variations. The control of the comfort criterion can

be achieved by minimizing a module of vertical body accelerations (which is a measure of exposure to fatigue resulted by vibration nuisance). However there is a problem of criteria conflict. While vehicle body acceleration is minimized the force wheel variation grows significantly.
The PZD control is possible in a given set of allowed friction forces:

$$T \in \Omega(V) := \{T_i \in R^2 \ : \ T_i = V_i c_i \ : c_i \in [c_{min}, c_{max}]\}. \tag{10}$$

Collection of allowed friction forces Ω (V) is shown schematically in Fig. 5 and is limited by a set of damping coefficients Cmin and C max dependent on the speed V. Damping coefficients depend on allowed voltage values of piezoelectric stack: Umin and Umax.

In comfort criterion the value of friction in any time is determined by minimizing an absolute value of the acceleration of a fixed point K - which corresponds (for example) to the position of the driver (cf. Fig. 4).

The value of this acceleration is described by the formula:

$$a_k = G^T \ddot{X}, \qquad G = [1, x_k]^T, \tag{11}$$

where the vector G determines the position of the point K.

On the basis of equation (2) the acceleration of the point K is described by following formula:

$$a_k = -G^T M^{-1} H(S+T) = a_0(t) + D^T T \ . \tag{12}$$

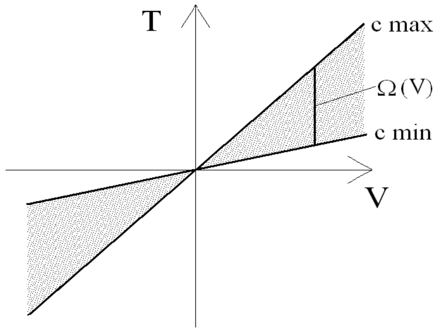

Fig. 5. Set of allowed solutions Ω (V)

where:

$$D^T := -G^T M^{-1} H, \quad a_0(t) := D^T S(U(t)), \tag{13}$$

Indicator of comfort is assumed and adopted in the form:

$$\kappa(T,t) = \left| a_o(t) + D^T T \right|, \tag{14}$$

The first criterion of minimizing vehicle body acceleration is the optimization problem of finding the best friction force in PZD. The problem is formulated in the following form:

$$T_K(t) \in \underset{T \in \Omega(V(t))}{Arg\ min} \left| a_o(t) + D^T T \right| , \tag{15}$$

where Arg min is the set of solutions, in which the minimized functional is convex but not strictly convex. A detailed description of optimization problem solution and control algorithm is described in [4].

The second criterion is reduction of vehicle wheel force variations $\Delta Q_i = T_i + S_i$. Indicator of reduction of wheel force variations is assumed and adopted in the form:

$$W(T,t) = \frac{1}{Q_{st}} \sqrt{\sum_{i=1}^{2} (\Delta Q_i)^2} = \frac{1}{Q_{st}} \sqrt{\sum_{i=1}^{2} (T_i + S_i)^2} \ , \tag{16}$$

where ΔQi (i = 1,2) - is the change in force under a single wheel of the vehicle and Qst=mg is so-called static value of wheel force.

The second criterion of reducing variation of wheel forces is also the optimization problem of finding the best friction force in PZD. The problem now is formulated in the following form:

$$T_N(t) = \underset{T \in \Omega(V(t))}{arg\ min} \left\{ \frac{1}{Q_{st}} \sqrt{\sum_{i=1}^{2} (T_i + S_i)^2} \right\}, \tag{17}$$

where arg min is the set of solutions, in which the minimized functional is strictly convex.

Double criterion can be obtained after combination of relationships (14) and (16) into one relationship allowing to find the best friction force in PZD. Effect of applying comfort and wheel force variations criteria on the friction force in PZD can be determined from the following relationship:

$$F(U(t),V(t)) := \alpha(t) \cdot T_K + (1 - \alpha(t)) \cdot T_N, \tag{18}$$

where α is the coefficient of weighted criterion.

Different control criteria can be selected on the basis of adjusting value of the weight coefficient α, which can vary in the range α [0,1]. Then, when the $\alpha = 1$ is chosen the criterion of comfort (15) is taken into account while for $\alpha = 0$ criterion of wheel force variations is applied (17).

6. Numerical simulation of the vehicle with PZD

In order to implement optimal selection of friction forces in damper described in the previous section a simplified vehicle model was developed in Matlab / Simulink [7].

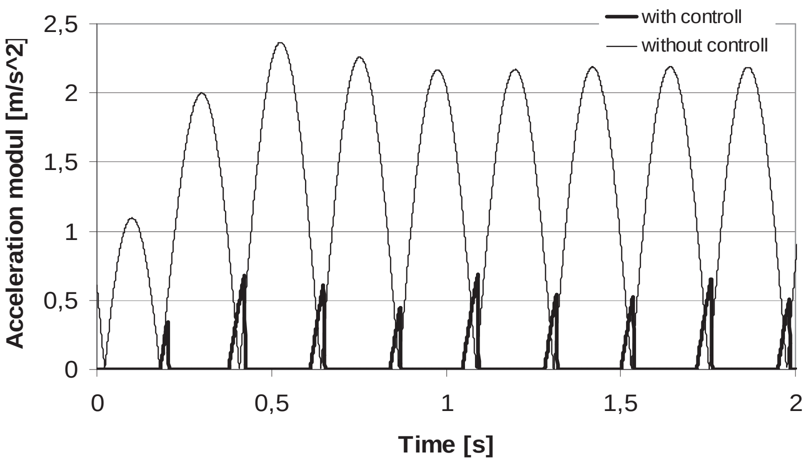

Fig. 6. Variations of absolute value of acceleration of point K with and without applied control algorithm

The following values of parameters were employed for the vehicle model in computer simulations: the oscillating weight m=1200 kg, spring stiffness k1=k2=47500 N / m, a1=a2=1.6 m and xk=0,.5 m. The model was tested with the control algorithm applied. In order to solve optimization problem it was necessary to define the limits of dimensionless damping factor γmin = 0,05 and γmax = 3 (respectively cmin = 750 Ns/m, cmax = 45000 Ns/m).

Simulation studies were carried out with harmonic sinusoidal kinematics extortion defined by formula:

$$\xi_i = \xi_0 sin(\omega t + \beta_i),\ \beta_1 = 0,\ \beta_2 = 2\pi\left(\frac{a_1 + a_2}{L}\right), \tag{19}$$

where L is wave length.

In Fig. 6 examples of numerical simulation results are presented. The comfort criterion was applied a=1. Kinematics extortion frequency was 2.25 Hz. To measure the effect of control comfort criterion was introduced coefficient on the basis of the international standard ISO-2631 criteria, where comfort, nuisance and annoyance are evaluated. Presented results show that it is possible to significantly increase comfort by decreasing values of acceleration amplitude. The exposure time in terms of comfort defined by ISO-2631 can be increased from 6 minutes (without control algorithm and with fixed value of dumping coefficient – original shock absorbers) to more than 9 h (with the controlled damping force with the comfort criterion algorithm).

7. Conclusions

We presented the results of numerical simulation and experimental investigations of a piezoelectric damper witch piezoelectric valve. Experimental studies were carried out to get PZD damper dissipation characteristics.

The results of numerical simulations of PZD devices were used to propose fast and efficient algorithm allowing improvements to passenger comfort. Proposed algorithm can be used to calculate appropriate value of signal to control in time prosperities of semi-active devices with respect of two opposing criteria: comfort (reduction of vertical accelerations) and safety (reduction of wheel force variations).

The presented results of vehicle simulation demonstrated that it is possible to significantly reduce vertical accelerations using controlled PZD dampers (shock absorbers). To measure the effect of control was imposed coefficient whose value in numerical simulations was significantly increased showing that it is possible to significantly extend time of comfort journey.

The proposed vehicle model and control algorithm can be used to simulate other controlled dampers like magnetorheological, electrorheological or electrohydraulic. It is also possible to link the model and algorithm with external systems such as MBS Adams.

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