TUNING TO ROAD AND LOAD PASSIVE SUSPENSIONS MULTI-MODELLING AND OPTIMISATION

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Abstract

This work explores the ability to apply multi-modeling technique of new suspension system based on a shock absorber model VZN (European Patent 1190184/20052). Here are presented the results of their own scientific research on the multi-modelling a auto vehicles suspension systems based on passive hydraulic shock absorbers with variable damping characteristics depending on the position of the sprung mass and road conditions. For such a system was proposed and verified by in Matlab-Simulink simulation, a procedure for optimizing the damping characteristics of the road conditions and load given. Suspension system is represented by a quarter-car multi-model with one degree of freedom and representative way perturbation by white noise. Proposed new criterion function in optimisation self-adaptive passive suspension.

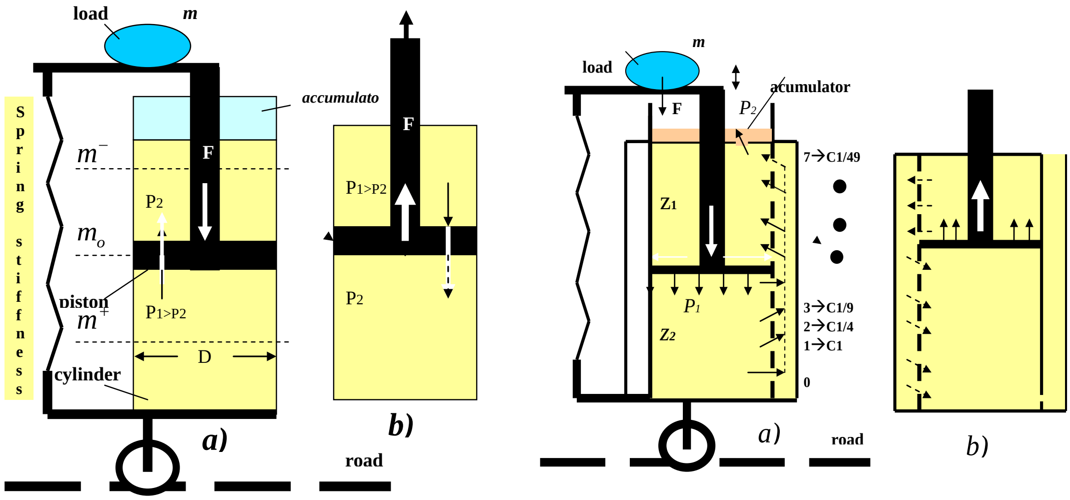
Keywords: *Road-suspension system simulation, optimality criterion, suspension multi-model.*

Introduction

Suspension systems are widely used in all types of land and air transportation as their means of protection against mechanical shock and vibration but they are also used for buildings, bridges, etc. for the same purpose of protection against jolts caused by various sources. There are three types of suspension systems: *active, semi-active* and *passive* First two types of suspensions haven't gained a widespread in the field of road vehicles and this was because as compared to the passive, are more complicated and have a very high cost, while at the same time presenting a low operational safety due to the existence of external sources power that, in case of failure, the entire system of suspension is put out of service. Active and semi-active systems arose from the need to automatic adjustment applied to the suspension to road and load conditions and involve excessive complication of the system by adding controllers, position sensors, actuators, etc. The main components are specific to any passive suspension spring stiffness and damper. This has led car manufacturers to continue to use the most ***classic passive dampers (CPD)***in vehicles passive suspension systems (Fig.1).The new concept of passive shock absorber ***VZN*** patented in Europe is a strong competitor of semi-active dampers and active suspensions (Niculescu, A.-I , European Patent 1190184/20052). VZN damper in the model "quarter car with a single degree of freedom (¼, 1D)" is shown in Figure 2 and has a structure just as simple as classic passive damper (CPD) and the same level of robustness and operational safety as CPD.(Niculescu, A., Dumitriu, D., Sireteanu, T., Alexandru, C ,2006).

The acronym VZN stands for “VARIABLE ZETA necessary for well NAVIGATION”, where ZETA represents the relative damping, which is changed stepwise automatically, according to the piston position.(Niculescu, A., Dumitriu, D., Sireteanu, T., Alexandru, C ,2008).. Most dampers used on road vehicles are hydraulic telescopic type (Dixon, J., 1999). In Figure 1 shows the simplified structure of a quarter-car with one degree of freedom model (1/4,1D)" of a passive suspension with telescopic shock absorber .Classic, suspension system model consisting of a cylinder which supports the wheel and piston which supports the load machine (chassis and traffic load) called sprung mass $\mathbf{m_1}$.

Changing the relative position of the piston to the vertical position due to changes wheel position u and change ***y*** position on the vertical of the



frame (ie. piston rod) called stroke S = y-u. The arch of suspension develops an elastic force ***Fs = kS*** (where k is the coefficient the elasticity of the arch) which balances the weight ***mg*** of the sprung mass. When the vehicle speed *v = 0 and*, the position of wheel u = 0, the piston is positioned at the appropriate level load average:

$$m_0 = \frac{m^+ + m^-}{2} \tag{1}$$

where m^+ is the maximum, and m^- is the minimum load (the chassis plus driver).The considered car has the following characteristics.These two values of *m*1 are , m^- = 600 [kg], m^+=1000[kg] considered in the simulations,*Smax*=0.236[m],*k*=114.085[kN/m].For CPD the damping coefficient values on compression stroke, C1c=823[Ns/m]=const and C1r=1993[Ns/m]=const ,on rebound stroke no.depending of the instantaneous piston position.

For the VZN shock absorber, the damping coefficient values increase on rebound stroke from 100 [N.s/m] up to 350000 [N.s/m] and on compression stroke from 200 [N.s/m] up to 600000 [N.s/m], i.e. more than 3000 times between minimal and maximal values, depending of the instantaneous piston position. (*A.I. Niculescu, C. Tabacu*, 2012).

The dampers are designed to dissipate energy from the vertical movement of the wheel and the ground. Energy dissipation is viscous friction of piston forced the oil to flow through holes made in the piston moving with relative velocity V. Damping force Fc = P_1A (where A is surface area on which the pressure P_1 acts) is negative in the race compression (CS) as in figure 1 On the return stroke (RS) oil is pushed through all the different orifice as shown in Figure 1b and piston develop positive damping force , Fr = P_1 (A-A_0), where A_0 is the area of the piston rod section.. For the CPD this damping force depends on the relative piston position during a stroke (CS or RS). To characterize the behavior of a classical damper using two kinds of features: force-displacement curve F (S) and force-speed curve. F (V). The authors showed in that, in the case of a single hole with *a* section area is developing a force F given by (C. Lupu, C. Tabacu, C. D. Câmpan, C. Eremia , 2013):

$$F(v) = \frac{qA^3}{2a^2} V^2 = C_1 V^2 \tag{2}$$

Where A is the area of the piston; ρ is density (for the incompressible oil);C1 is the damping coefficient of the damper with a single hole on each stroke. (Where n = 1). At CPD, the coefficient C1, called the damping coefficient remains constant for any position of the piston because during a stroke the area of hole "*a*" and A and density ρ have constant values in (2) . Damper-ideal curve F (V) is deformed in the actual case, because of the auxiliary phenomena such as, for example, oil temperature, undesirable leakage of oil through the gap between the piston and the cylinder wall changes the density and oil compressibility etc. Due to these factors, auxiliary damper curve obtained experimentally for the case of a single hole, is presented in Fig. 3 .The difference between the CPD with the holes in the piston and VZN shock absorbers is to produce a number of n> 1 holes, in the cylinder wall are arranged in line as shown in Figure 2 or the spiral. VZN damper wall in figure 2a and 2b have n = 8 hole with ***a*** section through which the speed of oil equal to the ***v*** The flows ***qi*** through each of the holes is thus equal to:

$$ai = av \qquad i = 1,2,....,n \tag{3}$$

In the case of CS, in accordance with the principle of continuity of flow, the amount of oil Q transferred per unit time from the space below the piston to the space above the piston (Fig. 2) is equal to:

$$Q = q_1 + q_2 + ... + q_n = AV \tag{4}$$

Where: A is the non-perforated piston, V=dS/dt is the relative velocity of the piston displacement. Whereas P2 = 0, it follows, in accordance with the law of Toricelli, the relationship between the pressure P_1 = F/A and the required kinetic energy transfer from the oil to the speed v in the Z_1-Z_2 through a hole section, the potential energy is equal to P_1-P_2 = P:

$$P = \frac{\rho v^2}{2} = F / A \quad \text{and} \quad v = \sqrt{2F / \rho A} \tag{5}$$

in that the density of oil, v is the velocity of the oil through the port, F is the force acting on the plunger rod (4) and (5) it follows:

$$F(V) = \frac{\rho A^3}{2n^2 a^2} V^2 = \frac{C_1}{n^2} V^2 \tag{6}$$

where: C_1 viscous friction coefficient ***n=1.***

From (4) it follows that for any value CPD pregnancy m^+ >m> m^- coefficient has not changed:

C = C_1 = const. The same is true when u wheel axle position changes between a maximum u^+(corresponding fluctuations in the profile of a very bad way) and a mean value u^0 (corresponding fluctuations in the profile of a good way) or when there are variations between a minimum (if a road considered very good).

Table 1 Autovehicle suspension Fuzzy control by changing the C1

Road & load conditions	C1	**Aim**
m1= **low**	**low**	To avoid collision to stroke ends
m1= **high**	**high**	To avoid collision to stroke ends
m1= **medium**	**medium**	The stability of movement
road=**bad**	**high**	To avoid collision to stroke ends
road =**good**	**medium**	The stability of movement
road =**very good**	**low**	The stability of movement (chassis&wheels)

Assessments of the road very bad, good or very good road may be assigned fuzzy representations that allow the design of fuzzy adaptive control algorithms to automatically control the damping force by controlling the amount of damping coefficient C of a semi-active damper in accordance with series of rules such as those of table 1 .Intuition, simulation analysis and experimental tests have shown that it can also implement adaptive control rules and a suspension system with shock absorbers type made VZN without any additional investment and maintaining the same level of safety..From table 1 ensues that by using the CPD absorbers there cann't be assured acceptable performances even if the road conditions remain stable that is why the requirements can become contradictory such as the case of a very good road to a maximal charge. In the case of VZN absorbers of suspension system these control rools of absorption rate are being memorized on the cilinder's wall by geometrical location adequate to an optimal number of openeings amongst which only some of them are active, regarding the piston's location. The reading of this mechanical memorizing for the adequate selection ot the absoption coefficient rate to the road conditions and charge is made through the adequate location on CS and Rs piston **.**The multi-modelling technique enables the modelling and analysis of nonlinear systems with time-varying parameters ,(like type suspension VZN).by classical methods.(S. Narendra ,J. Balakrishnan ,1997 and S. Narendra C., Xiang,2000).

In this paper several modeling technology is applied in representation VZN suspension model to simulate and optimize.(C. Tabacu, D. L. Câmpan, O. Dinu,2012 } .

CPD and VZN shock-absorbers Multi-Modelling(MM)

In most current scientific articles construction of passive suspension is treated as input-output linear dynamical system, CS and RS each race having different values of the parameter C1.

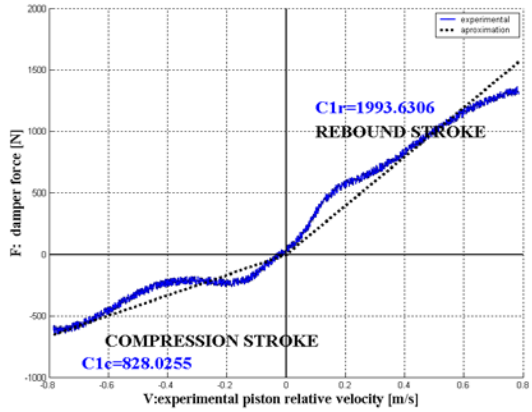


Fig.3 Experimental(*a*) and aproximation (b) damping curves of CPD for n=1 C1s=828Ns/m RS((C1r=1993Ns/m)

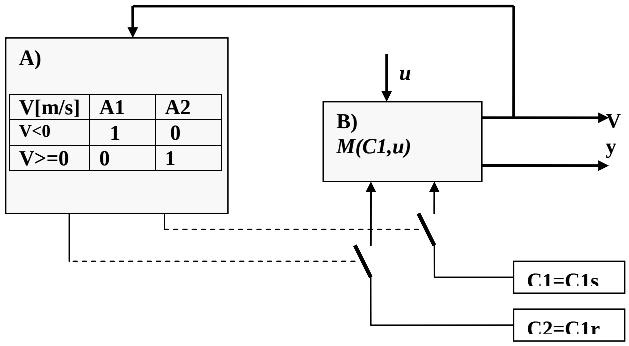


Fig. 4. The models base end commands base tor DACIA suspension multi-model

In Figure 3 presents the curve F (V) experimental and its linear approximation for CPD type this damper CS has valoareC1s = 823 [Ns / m] as less than C1r value = 1993 [Ns / m] on the return stroke RS. Since the model structure does not change but only change one of the parameter values , the system can be represented by two models similar in structure but differ in parameter values.The models ***M (C1s, t, u)*** and ***M (C1r, t, u)*** are called separately by assigning different parameter values C1, depending on the stroke Cs or RS. In Figure 4 is shown a block diagram of a multi-model ¼,1D, corresponding to that of the suspension from figure 1 , where y (t) is the output signal shape in time the sprung mass position and u (t) is input of MM that models the wheel axle position while the profile variations caused by the way .. According to the terminology proposed by the authors of MM technique , representing nonlinear systems contains a block called the commands base and another block B called the models base that communicate with each other as in figura.4.The structure of dinamic model ***M(C1,u)*** results from the balance of all forces acting on the model ¼, 1D , from figure1 :

- The linear behaviour of the dependence between the force and the displacement in case of flexible elements (*Fe=k(y-u))*;
- The linear dependence between the viscous friction force and the damper piston movement speed and the speed of the oil passing through the damper piston passing hole

$$F_{fv} = c(\frac{dy}{dt} - \frac{du}{dt}) = c(v1 - vo) = cV \;;$$

- The suspension mass m considered constant and focused on a point (centre of gravity) and the force of inertia ***Fi*** calculated as the product between the mass and the acceleration ($m\dfrac{d^2(y-u)}{dt^2}$);

From the equation of equilibrium of these forces, ***Fi+Ffv+Fe=0*** **,** resulting equation for the dynamic behaviour of the model ¼,1D of pasive suspension system:

$$m\frac{d^2y}{dt^2}+c\frac{d(y-u)}{dt}+k(y-u)=0 \tag{7}$$

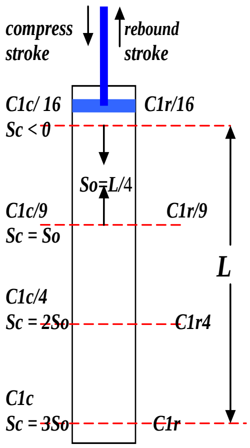


Fig.5 VZN damper with
4 holes / stroke

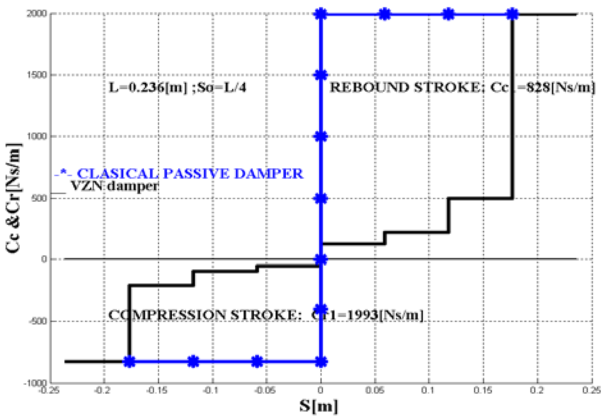


Fig.6 Cs(S) and Cr(S) steps Graphs forVZNdampe from Fig.5 and CPD type of damper from Fig.3

According to relation (6), where ¼ model, 1D supension system from Figure 2, which contains VZN shock absorber type, the coefficient C1 depends not only on the relative velocity V of the piston but depends and of S = y-u relative position of the piston in the a stroke CS or RS.Piston position is expressed by the number of the interval between two neighboring holes where the piston is positioned. If these ranges are practically zero length to obtain a continuous curve F (S) otherwise damping coefficients Cs and Cr changes in the step and just in the step change F (S) curve . For VZN damper in Figure 5 with only 4 holes arranged in line at distance So = L /3 to one another. Resulting graph in steps dependence of the damping Cs, according to Cr and S piston position for each stroke, as in Figure 6.

From Figure 6 it follows that PCD is constant CS and RS Heat factor and therefore can not satisfy the fuzzy control rules of Table 1To VZN damping coefficient is observed that self adapts to road and load function the allure graph C (S). From equation (6) it follows that along the optimal choice of the number n of holes and their arrangement on the cylinder wall is also can correct the graph C (S) and the choice of the area ***a*** of the holes.

The multi-model to the suspension system consisting of a spring stiffness and which is suspended mass VZN m VZN 4 was considered to be the lower hole vertically placed up on the cylinder wall at distance So each other. In this case, the resulting four intervals along a course in which they are active in a number of different holes (n = 1, n = 2, n = 3, n = 4) by the piston pushing the oil in the cylinder. This produces eight different values of the damping coefficient c selected according to the 8 rules of basic commands in comands base of Figure 7. The identical structures of dynamic models M1, M2, M3, M4 for suspension system shown in Figure 7 is equation(from models-base where,

$c \in \{C_{1r}, C_{1r}/4, C_{1r}/9, C_{1r}/16 \;\; for \;\; RS \;\; and \;\; C_{1c}, C_{1c}/4..C_{1c}/9, C_{1c}/16 \;\; for \;\; CS\}$

and other parameters: *m* and k - stiffness coefficient of spring coupled in parallel with VZN shock absorber for suspension mass *m*. The block diagram of multi-model built on relationships from commands base BC and the vehicle suspension dynamic model-base MB.

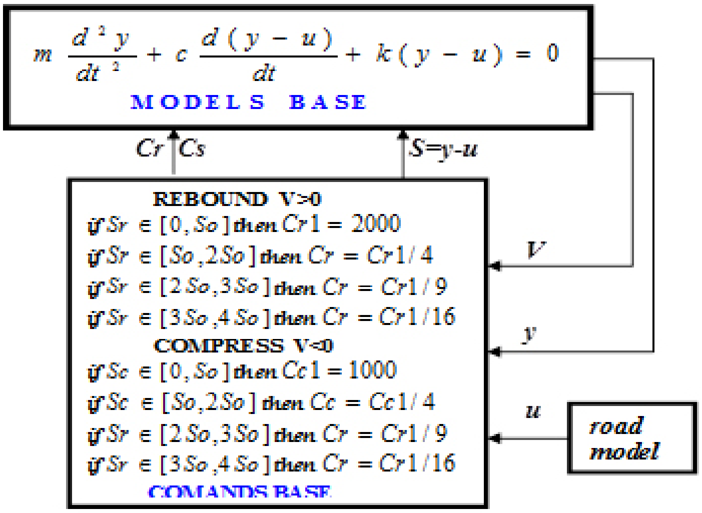
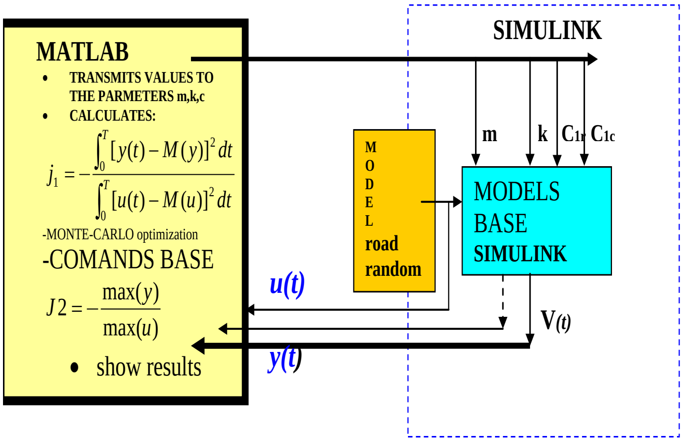


Fig 7 VZN based Suspension system , ¼ car 1Degree , multi-model

Each of the four models has the random perturbation input signal u (t) generated by road and has two output signals: *y* - position of the piston and *V*- relative velocity of the piston. Commands-base(CB) selects the proper viscous friction coefficient *c* and one of the four models according to the relative speed of the piston V(t), according to the rule from BC implemented on the CB for rebound and compress strokes. Each of the dynamic models M1, M2, M3, M4, seemed MB have identical structures [5], [9]. These structures are determined by the model chosen to represent the vehicle's suspension system. Traditionally, the road profile was modelled using some random processes .

Road-suspension system optimization

The simulation objective was to check the comfort and adaptability, random road conditions and load changes of a suspension system equipped with shock absorber VZN, compared with those of a suspension system equipped with CSD. In Fig. 5 is represented VZN damper with 4 holes for suspension system simulation using MM from the Fig. 7. To simulate suspension whose MM is shown in Fig. 7 was used MATLAB-Simulink. Base command was implemented in Matlab and the dynamic model was implemented in Simulink to simulate and road-induced noise in the system (Fig.8).



Comparison with DSA suspension provided with suspension system features a shock-absorber VZN was done by comparing the degree expressed by new two performance indicators J_1 and J_2. Indicator J_1 is expressed as the ratio of the variance of the output signal ***y*** of the suspension and variance of input signal ***u***:

$$J_1 = \frac{\int_0^T [y(t)-M(y)]^2 dt}{\int_0^T [u(t)-M(u)]^2 dt} = \frac{\operatorname{var}(y)}{\operatorname{var}(u)}$$

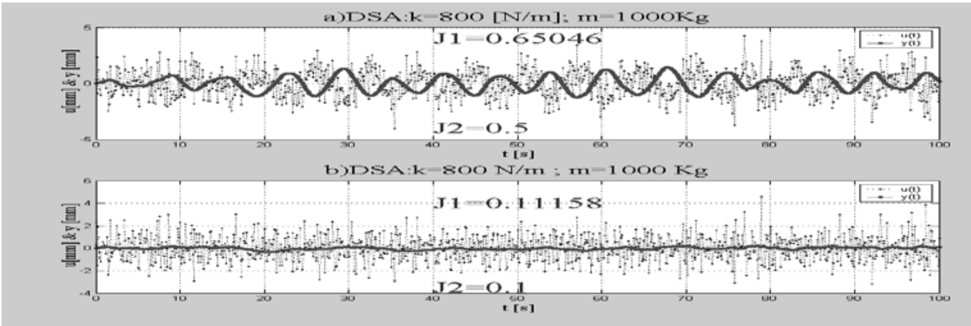
where M (y(t)) and M (u (t)) are the averages of the signals .

J_2 indicator is expressed as the ratio of the maximum amplitudes of the two signals:

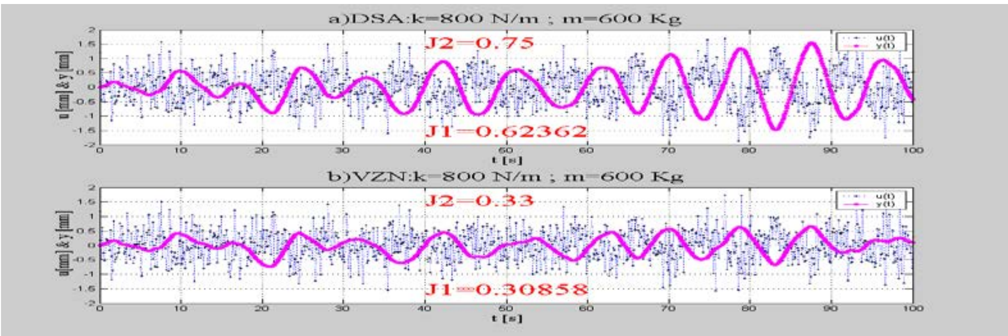
$$J_2 = -\frac{\max(y)}{\max(u)} \qquad (9)$$

If the suspension system is stable, in the extreme case, $J_1 = J_2 = 1$ then the system no filtering the noise and var (y) = var (u) and max (y) = max (u). Ideally the filter is the total $J_1 = J_2 = 0$.

Calculation and display of these indicators and simulation the suspension systems VZN and CPD is done by Matlab cases based on the signals y and u received from Simulink(Fig.8).



Cases 1a CPD **& 1b VZN:** m=1000 Kg & max(u)=5



Cases 2a CPD **& 2bVZN:** m=600 Kg & max(u)=2

Figures 9 presents the time evolution of the signals ***y*** and ***u*** in two different situations of the two systems simulated. We chose these two different road and task to check that the VZN suspension system helps to adapt to changes in road conditions and load.

Table 2 Simulation results

Figure	**Damper**	**M [Kg]**	**Max (u)** [mm]	J_1	J_2
9a Case 1	**CPD**	1000	5	0.6	0.6
9b Case 1	**VZN**	1000	5	0.1<0.6	0.1
9a Case 2	**CPD**	600	2	0.6	0.6
9b Case 2	**VZN**	600	2	0.3<0.6	0.3

Table 2 presents the results obtained in the two test cases with DSA suspension systems and shock-absorber VZN. The table shows that adapts VZN suspension with 4-5 times better to changing road conditions and load..

Conclusions

Analysis of simulation results showed substantial increase yield by the ratio of the signals u and y variants of DSA than five times the compression stroke compared. The simulations indicated that the multi-model was quite effective over wide ranges of unmeasured disturbances and process changes. Analysis of simulation results showed a substantial increase of the ratio J1 of variances yield signals u and y of CPD compression standard over in stroke compared with stroke rebound. .

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