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Design and Evaluation of a Suspension System with Variable Parameters

Shadi Munshi, Sufyan Azam, Mohammad Gadoori

Abstract*: *Suspension systems are designed for the increase in comfort and stability in vehicles while driving. Parameter changes in these systems affect overall performance. Researchers are continuously working on the performance enhancement of suspension systems by designing dampers of variable damping coefficient. In this research work a quarter car model suspension system was developed to demonstrate experimentally the influence of changing the damping coefficient, the stiffness, and the loading level to fully understand the allowable combination of parameters for a smoother ride. A variety of different test scenarios are implemented to the system to observe the variations in damping ratio. The damping ratio of the system found to be dependent on the design of the opening mechanism.The valve opening methods can give different results with the same parameters.

Keywords*: *variable damping, suspension, stiffness, stability, comfort.

I. INTRODUCTION

The automobile suspension is the system of devices which support the vehicle body on the axle. The vehicle body itself refers to the sprung mass, which consists of the housing as well as its contents, the engine and some other mechanical parts. On the other hand, the axle refers to the un-sprung masses, which consist of the wheels and tires. Finally, the suspension system consists of springs, dampers and actuators. In order to improve the overall performance of automotive vehicles, suspension system has the ability to isolate the wheel of the car from the car body in case of any road disturbance that may occur. The purpose of suspension system is to improve the ride comfort, road handling and stability of vehicles. Basically, suspension systems can be Passive, Semi Active or Active depending on the components used. Although an active suspension provides better performance than semi-active suspension, it has major disadvantages such as the need for a large external power source, increase complexity, cost and decrease reliability. And therefore, they are not commonly used in commercial vehicles. Issues related to the design and control aspects in active suspension systems appear to be the real challenges. A semi-active suspension system combines the advantages of both active and passive suspensions providing good system performance compared to passive suspensions. It is economical, safe and does not require either higher power actuator or a large power supply.

Different methods and experiments have been used by many researchers in order to improve the performance of vehicles ride, comfort, road handling and stability. K. Kamalakannan, A. Ela-yaPerumal, S. Mangalaramanan, K. Arunachalam [1] simulated and analyzed a simple low-cost semi active suspension system using "MATLAB and SIMULINK" platform and establish its superiority, and also involves the development and simulation of a virtual quarter car model. Galal Ali Hassan [2] studied the effect of suspension damping and car speed via examine and analysis the dynamic of car passing a circular hump which is performing by using a quarter car model and MATLAB software. Abroon Jamal Qazi, Afzal Khan, M. Tahir Khan, Sahar Noor [3] modelled the system using quarter car model in Simulink software. Three different damping coefficients were selected in order to analyze the response of semi-active suspension system in terms of ride comfort. The fuzzy logic controller was modelled using Fuzzy Tool Box. S. Segla and S. Reich [4] optimized and compared the performances of the different suspension systems (quarter car models) and shown, that not only the active, but also semi-active suspension system is able to improve the ride comfort significantly compared with the classical passive suspension systems by using the relaxation damping, vibration absorber, and the additional dampers between the engine and axle. Abroon Jamal Qazi, Umar A. Farooqui, Afzal Khan, M. Taher Khan, Farrukh Mazhar, Ali Fiaz [5] designed various models on the basis of various control algorithms and compared their response in terms of road handling and ride comfort. The quarter car passive and semi-active suspension systems were modelled in Simulink software. The inputs and output of optimized fuzzy logic controller were normalized and gain factors were incorporated in the system. Off-line tuning method was performed in order to evaluate the gain factors using Particle swarm optimization (PSO) technique. Based on the optimized parameters, the maximum output of the damper is selected. Carlos A. Vivas-Lopez, Diana Hernandez-Alcantara, Manh-Quan Nguyen,

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Ruben Morales-Menedez, and Olivier Sename [6] proposed a new semi-active control that includes a Linear Parameter Varying (LPV) controller designed to improve the ride comfort incorporating a Force Control System (FCS) to transform the force command from the LPV controller to an input signal for the Electro-Rheological (ER) semi-active damper. Then the semi-active control system was evaluated in a Quarter of Vehicle model under two tests: Bump and Road Profile. Bana Kasemi, Asan G. A Muthalif, M. Mahbubur Rashid, Sharmila Fathima [7] conducted an experiential in order to establish the behaviour of the Magnetorheological (MR) damper, the behaviour of (MR) damper was studied and used in implementation vibration control (FUZZY-PID Controller). Juan C. Tudon-Martinez, Diana Hernandez-Alcantara, Ruben Morales-Menendez [8] proposed a novel control Linear Parameter Varying (LPV) for an automotive semi-active suspension based on a quarter vehicle model and compared the results to systems associated with an uncontrolled damper and a standard LPV controller without mass adaption. M. Witters, J.Swevers [9] assessed the dynamic damper behaviour by proposed a black box simulation model structure using continuously variable, electro- hydraulic semi-active damper (CVEHSA), A neural network based output error (NNOE) and performed all measurements on a position controlled hydraulic test rig. The control system technique was discussed by Jiangtao Cao, Student Member, IEEE, Honghai Liu, Senior Member, IEEE, Ping Li, and David J. Brown [10] through reviewed computational intelligence involving approaches in active vehicle suspension control systems focusing on the problems raised in practical implementations by their nonlinear and uncertain properties. They explored the state of the art in fuzzy inference systems, neural networks, genetic algorithms, and their combination for suspension control issues. Li Sui, Zhen Wang, Geng-Chen Shi, Guo-Zhong Li [11] developed the variable stiffness suspension by used the adjustable length of coil or leaf springs. T. Ram Mohan Rao, G. Venkata Rao, k.Sreenivasa Rao & A. Purushottam [12] conducted a comprehensive analysis of novel hybrid semi-active control algorithms and compared the semi-active and passive systems in terms of human body vibrational displacements and accelerations. M.D. Symans, M.C. Constantinou, D.P. Taylor, and K.D. Garnjost [13] developed a semi-active fluid viscous damper for use as supplemental seismic energy dissipation devices which used in structure aspects and eliminate the effectiveness in the event of earthquake incidents. In the present study, a semi-active suspension system “prototype model” is designed to study the performance of the developed system in terms of sprung mass displacement. The system was designed using SolidWorks and results were validated by Matlab / Simulink. The developed prototype model has a variable damper, springs with different constants and variable dead weights.

II. SYSTEM DESCRIPTION

A. Working Principle

The quarter car vehicle model equipped with a semi-active suspension system was taken into consideration in the prototype model with validation using calculations. This model consists of (sprung mass - mb) which represent the vehicle body and (un-sprung mass - mw) which represent the wheels and suspension elements (spring and semi-active damper). The road disturbance profile was provided through a pneumatic actuator along with distance sensors in order to measure and evaluate the system behavior. The developed system consists of the following components:

- Main Housing Body
- Pneumatic Cylinder
- Variable Damper Assembly
- DC Servo Motor
- Linear Potentiometer
- Springs and Dead Weights

The system model is divided into three zones as shown in Fig1. The bottom zone which consists of a pneumatic cylinder attached to the main housing body, responsible to provide a required pressure that enables the actuation of the middle zone to simulate a road bump as a step input. The middle zone consists of a variable damper assembly and a DC servo motor responsible to modulate the damping coefficient parameter. The upper zone is the body with variable weights and can represent the system responses. Two linear potentiometer sensors are used to measure the step input and system responses respectively.

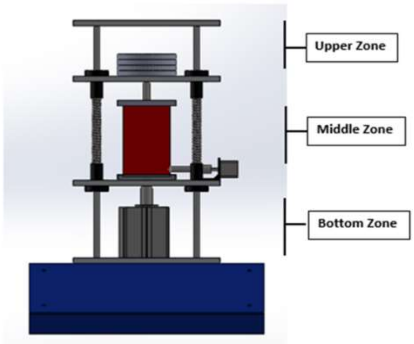


Fig1:CAD Model of the suspension system

B. Variable Damper Design

The variable damper consists of cylinder house and piston cylinder. The oil flows inside the piston cylinder through the two openings (bottom and upper) on the body of piston cylinder. Fig2 shows the bottom hole of piston cylinder that is aligned horizontally with the bottom hole of cylinder housing, this kind of action is designed to modulate the oil flow inside the system by controlling the DC servo motor shaft through the cylinder housing choking the flow of oil thus changing the damping coefficient of the system.

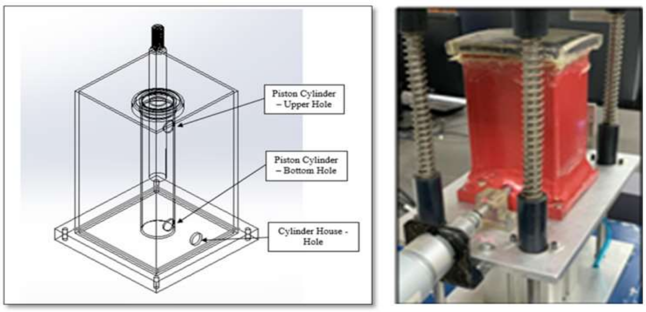


Fig2: Variable damper Design

III. EXPERIMENTAL SETUP

The experimental apparatus, as shown in Fig3, consists of the following:

- Air Regulator
- Pneumatic Cylinder
- Bottom Linear Potentiometer
- Variable Damper Assembly
- DC Servo Motor
- Upper Linear Potentiometer

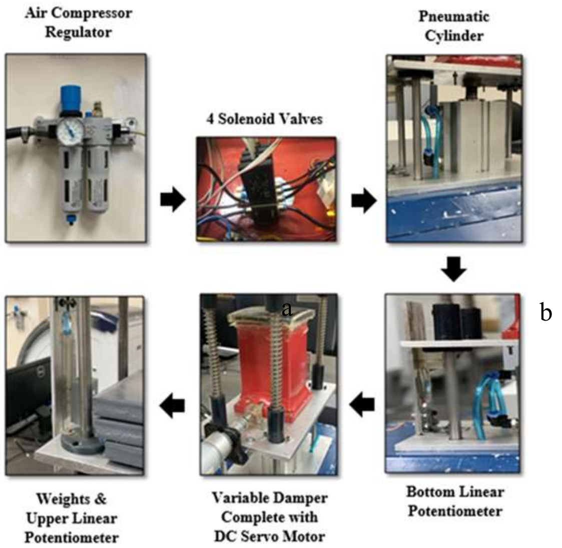


Fig3: Experimental apparatus flow

A. Methods

The step input response which is generated by the air compressor system is measured through the bottom linear potentiometer and the variable damper assembly with seven different positions of valve opening are considered during the experimental process. Also, two different stiffness coefficients with different weights are applied to validate the performance of the system response which is measured by a linear potentiometer. The step input and system responses were analyzed on OriginPRO software to generate plotted graphs, furthermore, the 2nd order system log decrement method [14] is used to identify and extract the parameters of the underdamped cases in time domain.

The log decrement $\delta = \frac{1}{n}\ln[\frac{y_i - y_f}{y_{i+n} - y_f}]$ is calculated to get the damping ratio ζ.

Whereas, $\zeta = \frac{\delta}{\sqrt{\delta^2 + 4\pi^2}}$

The damping ratio can be used to find the natural frequency w_n of vibration of the system from the damped natural frequency w_d [14].

$$w_d = \frac{2\pi}{T}$$

T is the time between two successive amplitude peaks of the underdamped system.

Then, $w_n = \frac{w_d}{\sqrt{1-\zeta^2}}$

From the above parameters, the damping coefficient and critical damping coefficient can be found by applied the followings formula, [15].

$$C = 2\zeta m w_n$$

$$C_c = \frac{C}{\zeta}$$

For result confirmation, a MATLAB transfer function used with experimental extracted data plotted along with simulated responses for comparison purposes.

$$y(t) = y_o + (y_f - y_o)[1 - \frac{\exp(-\zeta w_n t)}{\sqrt{1-\zeta^2}}.\sin(w_d t + \phi)]$$

The Table 1 shows variable parameters used in this study. Combinations of some parameters from Table 1 were studied with different cases as shown in Table2.

Table 1. Variable input parameters

I. Variable Damper - Valve opening (mm)							
1	2	3	4	5	6	7	8
II. Variable Weights (Kg)							
3.136	5.506	7.871	10.31	12.74	15.04		
III. Variable Stiffness Coefficients (N/mm)							
K1 = 1230				K2 = 1590			

Table 2. Selected cases applied to the suspension system

#	Valve Opening (mm)	Stiffness Coefficient (N/mm)	Weights (Kg)
Case 1	3	K1	5.506
Case 2	3	K1	7.871
Case 3	3	K1	10.31
Case 4	5	K1	5.506
Case 5	5	K1	7.871
Case 6	5	K1	10.31
Case 7	8	K1	5.506
Case 8	8	K1	7.871
Case 9	8	K1	10.31
Case 10	3	K2	5.506
Case 11	3	K2	7.871
Case 12	3	K2	10.31
Case 13	5	K2	5.506
Case 14	5	K2	7.871
Case 15	5	K2	10.31
Case 16	8	K2	5.506
Case 17	8	K2	7.871
Case 18	8	K2	10.31

IV. RESULTS AND DISCUSSIONS

To gain an overview of the system profile and parameter variations, experiments were performed as follows for a predefined bump profile, a unit step input, to the system. The stiffness coefficient was set to K1 with a valve opening of 3mm while changing the loads 5.506kg, 7.871kg and 10.31kg respectively. The previous step was repeated with valve openings of 5mm and 8mm respectively. For the second set of data, the stiffness coefficient was set to K2 and the previous experiments were repeated. The experimental data was then compared to a set of calculated data to verify the results. Fig4-6 display the experimental and calculated responses of cases 1 to 3. In each case, the system response was obtained experimentally and the damping coefficient was calculated and compared to the theoretical one. Table 3 shows steady state response of the suspension system modelled in MATLAB after extracting parameters from the actual system response using log decrement method. The rise time Tr of the system in case 3 is exceptionally slow as compared with the other cases, it's because of high sprung mass with small opening.

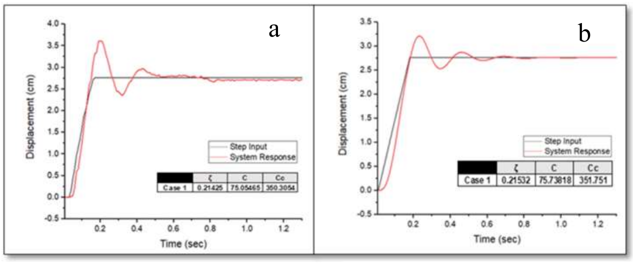


Fig4. Experimental and calculated response for case1

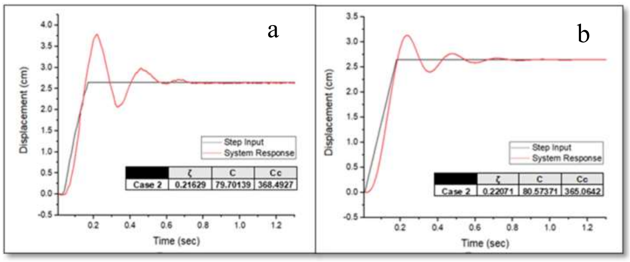


Fig5. Experimental and calculated response for case2

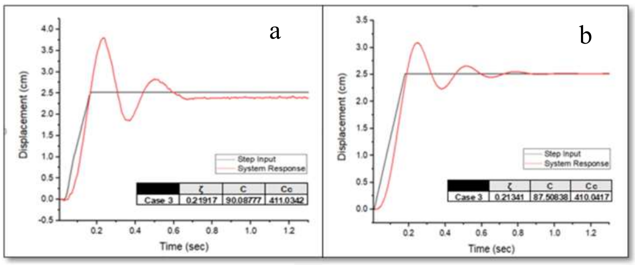


Fig6. Experimental and calculated response for case3

Table 3. Calculated Steady-State Response of the System from the selected cases

#	T_r (sec)	% Overshoot	$\Delta Y/\Delta T$ (cm/sec)
Case 1	0.0742	30.9	2.77
Case 2	0.0755	43.1	3.32
Case 3	0.10257	50.8	3.5
Case 4	0.0674	37.661	3.216
Case 5	0.0732	50.945	2.832
Case 6	0.0776	59.327	2.902
Case 7	0.071	42.105	2.67
Case 8	0.0695	60.203	2.52
Case 9	0.0726	58.170	2.33
Case 10	Overdamped	N/A	N/A
Case 11	Overdamped	N/A	N/A
Case 12	Overdamped	N/A	N/A
Case 13	0.0709	29.95	3.042
Case 14	0.0733	44.67	3.29
Case 15	0.073	54.52	2.944
Case 16	0.0826	33.015	3.214
Case 17	0.0677	50.295	2.891
Case 18	0.0715	59.69	2.825

In Fig7, the system damping ratio is shown against constant loading but variable valve opening for the constant spring K1. The openings 2-4 show a sharp decrease in the damping ratio for the weights W1, W2 and W6 although these weights do not show much change in the damping ratio for the later openings. The middleweights W3, W4, and W5 showing oscillating behaviour of the damping ratio at full valve opening range. These weights do not show many variations in the damping ratio as compared to the weight W1 that gives 0.65 down to nearly 0.3 damping ratio values.

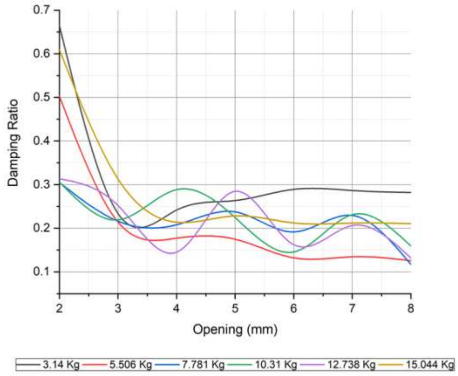
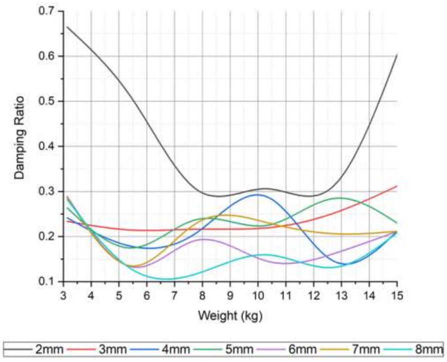


Fig7. Damping ratio variations at constant weights with the variable openings for spring constant K1

Fig8 shows the damping ratio variations at the constant opening, but variable sprung weights for the constant spring K1. This graph shows how at a certain opening, the different weights affect the system damping ratio. The 2mm valve opening, V2, gives a high damping ratio at the low and high loads. It can be seen from the graph, the damping ratio at this opening for different weights behaves way different as compared to the other openings. At the 2mm opening, the first three weights W1, W2 and W3 show a sharp decrease in the damping ratio while the same opening gives a sharp increase in damping ratio between the weights W5 and W6. But between W3 to W5 system damping ratio is almost constant at the first valve opening. As seen from the graph, the 3mm opening i.e., V3 gives a smooth increase in the damping ratio as compared to the other openings that show an oscillating damping ratio behavior at different loads. The minimum damping ratio is about 0.1 and can be seen at 8mm



valve opening.

Fig8. Damping ratio variations at a constant opening with the variable weights for spring constant K1

Fig9, shows the system damping ratio against constant loading but variable valve opening for the constant spring K2. In this graph, more stiffness is added due to the change in the range of weights in the system. The damping ratio at 3mm opening for 5.506Kg is lower in fig.9 as compared to the fig. 7 whereas at 8mm opening damping ratios are almost the same for the above weight.

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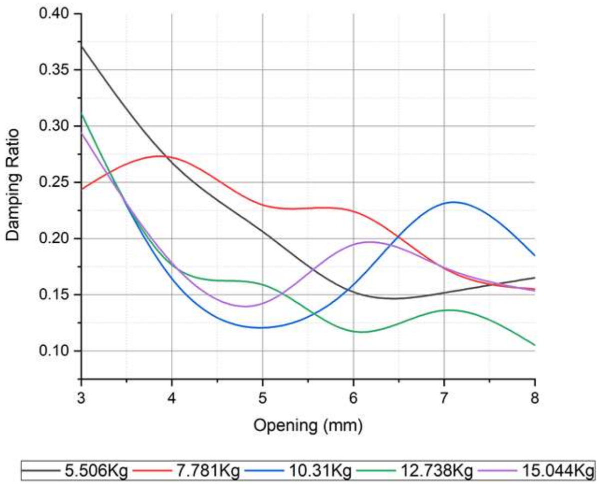

Fig9.Damping ratio variations at constant weights with the variable openings for spring constant K2

Fig10 shows the damping ratio variations at constant openings, but variable sprung weights for the constant spring K2. In this case, the damping ratio for the 3mm opening is higher than that for spring constant K1. The overall damping ratio is ranged from 0.1 to 0.3 for the openings from 4mm to 8mm for both spring constants K1 and K2.

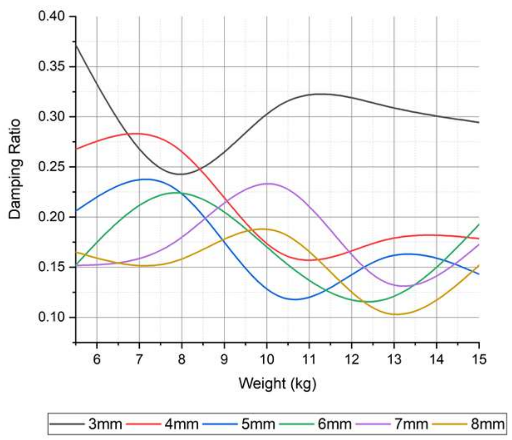

Fig10. Damping ratio variations at a constant opening with the variable weights for spring constant K2

Fig11 shows percentage overshoot against different valve openings at constant weights for spring constants K1 and K2 respectively. The system shows higher %age overshoots with heavy weights. For the spring constant K1, 3mm opening shows more overshoot range as compared to K2 with the same parameters.

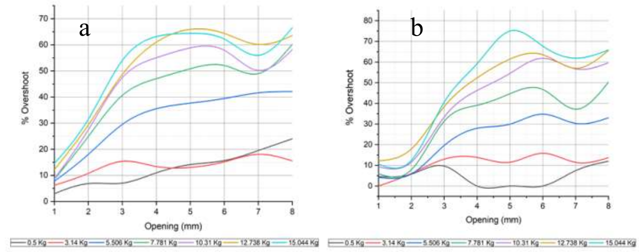

Fig11. Overshoot variations at constant weights with variable opening for spring constant K1 and K2 respectively

In Figure 12, the data was plotted for variable weights at constant opening with spring constants K1 and K2. These plots show the percentage overshoot of the system against constant openings and variable weights. The percentage overshoot range for K2 is more for 4-8mm openings as compared to K1 for the same openings whereas this behavior is observed opposite for the openings 1-3mm.

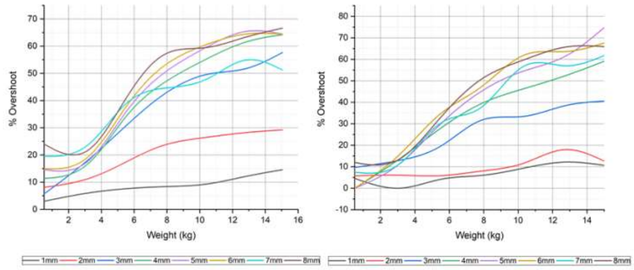

Fig12. Overshoot variations at a constant opening with variable weight for spring constant K1

V. CONCLUSIONS

Experimental and validated results of the damping ratio have been presented in this paper. Different types of compression springs, weights, and variable damping with different valve openings have been tested in the prototype model to validate the performance of the system. It has been observed by the system behavior that changing system parameters e.g., weights, valve openings and spring constants give different system response. The damping ratio of the system is dependent on the design of the opening mechanism; different styles of the valve opening can give different results with the same parameters. An ongoing work is carried out, where different input profiles will be added to the system to see system response at various input speeds.

ACKNOWLEDGMENT

It is optional. The preferred spelling of the word “acknowledgment” in American English is without an “e” after the “g.” Use the singular heading even if you have many acknowledgments. Avoid expressions such as “One of us (S.B.A.) would like to thank ... .” Instead, write “F. A. Author thanks ” *Sponsor and financial support acknowledgments are placed in the unnumbered footnote on the first page.*

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