STUDYING THE EFFICACY OF ROAD DISTURBANCE AMPLITUDE AND SPEED INCREASING ON PERFORMANCE OF SEMI ACTIVE SUSPENSION SYSTEM

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ABSTRACT

In this paper the performance of designed semi active suspension for a passenger car has been investigated under different road condition and different vehicle speeds. Semi active suspension is an appropriate scheme for vibration control in vehicles due to its high performance, which is close to active suspensions. However semi active suspension has less cost and energy usage than active suspension. Two most significant parameters in designing process of each suspension system are ride quality and handling parameters of the vehicle, which have been used for studying the performance of suspension system. Transferred chassis acceleration to the passenger has been assumed as a ride quality and tire deflection has been assumed as handling of vehicle. Parameters of the Renault Megane Coupe have been used for modelling and simulation of a vehicle moving at speeds of 80, 108 and 130 [Km/hr] in a rough road with disturbance amplitudes of 0.02, 0.05 and 0.08 [m]. For investigating the performance of semi active suspension system a quarter car model with two degrees of freedom has been used. Performance of the semi active suspension system has been investigated in the range of damping coefficients from 1000 to 6000 [N.s/m]. The obtained results show that by increasing the vehicle speed and road disturbance amplitude, transferred acceleration to the passenger and tire deflection will be increased. This means that ride quality and handling of the vehicle have been decreased. Also the range of 1000-2000 [N.s/m] for damping coefficient and disturbance amplitude domain from 0-0.06 [m] is the most appropriate amount due to its low rate of sprung mass acceleration and tire deflection.

KEYWORDS: *semi active suspension system, ride quality, road holding, passive suspension system, quarter car model.*

I. INTRODUCTION

The fundamental goals of a suspension system are to support the vehicle weight, to isolate the vehicle body from road disturbances, and to maintain the traction force between the tire and the road surface. Suspension systems composed of spring-type elements in parallel with dissipative elements. Suspension springs provide comfort isolation for the vehicle passengers, and help maintaining contact pressure between the tires and the road surface. Dampers are used to dissipate energy from suspension motions, operating by the viscous action of oil as it is forced through small orifices within the damper. [1-3]. The Design of the suspension (i.e. the choice of the most suitable values of the stiffness and damping) is concerned with (i) the car body acceleration for ride comfort, (ii) the tire deflection for road holding and (iii) the suspension travel (rattle space) which must remain within fixed limits. An ideal suspension would minimize these three quantities for any road and operating condition, which is not achievable for a suspension having constant stiffness and damping [4-6]. Spring rate and damping are chosen according to comfort, road holding and handling specifications.
A suspension unit ought to be able to reduce chassis acceleration which is related to ride and comfort. Numerous studies [7-10] have been conducted on the description and improvement of ride comfort.

To this end parameters, indicative of ride comfort (e.g. vertical acceleration, absorbed power), have been described and levels of acceptance laid down in standards such as ISO 2631 [11], BS 6841 [12] and VDI 2057 [13]. To improve the ride quality, it is important to isolate the sprung mass (the car body) from the road disturbances and, particularly, to suppress the vertical vibrations near 5 Hz (4-8 Hz), which is a sensitive frequency range to the human body (and the lateral vibrations at 1-2 Hz according to ISO 2631. Vehicle's suspension system can be broadly classified in to the three categories [7, 10, 14, 15] as follows.
In this paper, three types of suspension system have been clarified and a 2DOF quarter car model has been presented within governing equations. Finally the results have been presented and discussed.

1.1. Passive Suspension

A passive vibration isolation system is the simplest way to protect a dynamical system from vibration inputs. Generally speaking, this system involves a parallel mounting of a spring as an energy storing element and damper as an energy dissipating element. Passive suspension is linear in nature. It is based on the principle of energy dissipation by the damper and does not require an external power source for operation and utilizes the motion of the structure to develop the control forces [16, 17]. This system has the advantages of simplicity, low cost, being easy to manufacture, implement and maintain. In this system, vibration isolation is accomplished through the insertion of a linear stiffness element and a linear damping element between the vibration source and the system requiring protection. In the case of a passive suspension, the stiffness and damping element characteristics, namely k and C cannot b varied once chosen. Hence, it is necessary to choose these components carefully to provide the best possible performance [18]. However, this choice involves a number of compromises arising from the desire that a suspension must appear soft to minimize acceleration levels and simultaneously hard to control vehicle attitude changes and maintain good tire/ground contact [10, 19]. Passive suspensions have inherent limitations as a consequence of the trade-off in the choice of the spring rate and damping characteristics, in order to achieve acceptable behaviour over the whole range of working frequencies. Chalasani [20] has demonstrated that increasing the passive suspension damping coefficient enhances vehicle comfort. Woodrooffe [21], Cole and Cebon [22] and Cebon [23] examined the passive suspension design of a heavy vehicle to minimize road damage. Zehsaz et al. [24] reduced the transmitted vibrations of tractor cabin through optimization of passive suspension parameters via both experimental method and Finite Element modelling. Morales et al. [25] proposed a method to reduce the vibration of unbalanced machinery using an adaptive-passive magnetoelastic suspension.

1.2. Active suspension

Active suspension can greatly improve vibration isolation performance compared to passive suspension [26]. In an active suspension the interaction between vehicle body and wheel is regulated by an actuator with variable length. Hence A first choice in the design of a fully active suspension is the type of actuation. Active suspension employs pneumatic, hydraulic, hydro pneumatic, piezoelectric and electromagnetic actuator. Actuator which in turn create desired force in the suspension system and applies between body and wheel a force that represents the control action generally determined with an optimization procedure. Active suspension require an energy source (such as a compressor or pump), sensors, controllers, actuators, servo-valves, switching devices and a computer control in order to achieve superior vibration isolation. Active suspensions may consume large amounts of energy in providing the control force. In consequence they are more expensive, more complex and less reliable and so the implementation of active shock and vibration isolation systems has been limited [27, 28]. Different optimal control techniques like linear quadratic regulator (LQR), linear quadratic Gaussian (LQG) control, fuzzy logic and neural network methods have been used in the area of active suspensions [9]. Akçay et al. [29] derived the suspension travel and the tire deflection for a quarter-car active suspension system using the vertical acceleration. In addition they studied multi-objective control of a half-car active suspension system using linear matrix inequalities [30]. Gao et al. [31] investigated active seat suspension system via dynamic output feedback control. Tunga et al. [32] proposed an active suspension mechanism for 3DOF twine-shaft vehicles using exponential decay control and particle swarm optimization (PSO) techniques. They used PSO method to solve optimization problem. Chen et al. [33] tried to provide a systematic probe into necessity of the active suspension system based on LQG control for supplying some reference to optimal suspension design.

Sande et al. [34] studied the control of novel electromagnetic active suspension system for quarter car model in both simulation and experiments.

1.3. Semi active suspension

Semi-active control devices offer reliability comparable to that of passive devices, there's yet maintaining the versatility and adaptability of fully active systems, without requiring large power sources [14]. The main advantage of this system is to adjust the damping of the suspension system without any use of actuators. In a semi-active suspension the amount of damping can be tuned in real time. Hence most semi-active devices produce only a modulation of the damping forces in the controlled system according to the control strategy employed. In contrast to active control devices, semi-active control devices cannot inject mechanical energy into the controlled system and, therefore, they do not have the potential to destabilize it. The kernel of a semi-active system is the controllable damper. A wide range of dampers exist based on a variety of dissipating mechanisms (deformation of viscoelastic solids, throttling of fluids, frictional sliding, yielding of metals, and so forth). The following is a list of some common types of dampers employed in engineering applications [14].

- Viscous dampers
- Viscoelastic dampers
- Friction dampers
- Magnetorheological fluid dampers
- Electrorheological fluid dampers
- Shape memory alloy dampers
- Tuned mass dampers
- Tuned liquid dampers

Details on the physical principles of these dampers can be found in [14, 26]. Semi-active suspensions were firstly introduced in the 1970s by Crosby and Karnopp [35] and Karnopp et al. [36]. Similar work was performed by Alanoly and Sankar in terms of active and semi active isolators [37]. Liu et al. [38] studied four different semi-active control strategies based on the skyhook and balance control strategies and compared them with an adaptive-passive damping strategy. Leluzzi et al. [39] designed the control strategy so developed process and performance of a semi active suspension system for a heavy truck. Liu et al. [40] proposed theoretical and experimental analysis of a new configuration using two controllable dampers and two constant springs for semi active suspension. Marcu [41] used the Magneto-Rheological (MR) damper in a class 8 semi-truck cab semi active suspension system to improve ride quality. He implemented designed controller called Hierarchical Semi Active Control (HSAC). Spelta et al. [42] implemented a control system via a semi active electro-hydraulic damper for a semi active suspension in a 2wheel vehicle and analysed the experimental results. Shu et al. [43] studied a 7DOF full-body dynamic model of vehicle semi active suspension using a double-loop control, identified results show effectiveness of proposed method. Collette et al. [6] used a quarter car model to investigate the effect of unintended high frequency excitation produced by the semi active sky-hook control on isolation of suspension system. Buckner et al. [44] used a multi-objective genetic algorithm (MOGA) to evaluate the optimization of control algorithms for semi active vehicle suspensions. Poussat-Vassel et al. [45] proposed an overview of some semi active suspension control strategy to evaluate them and applied to various control approaches.

II. SIMULATION SETUP

A quarter car model subjected to harmonic road excitations with two degrees of freedom has been used in this study. In order to simulate a quarter car model with two degrees of freedom using MATLAB, second order movement equations for both sprung mass and unsprung mass has been identified. Basically the system is subjected to harmonic excitation. Applied excitation to the system is in the form of displacement, which is defined as $A_0 \sin\omega t$ where A_0 is the amplitude of the road disturbance, ω is the excited frequency and t is the time. The velocity and acceleration vectors of sprung mass and unsprung mass defined based on displacement vectors. Finally using Cramer's rule and implementing real quantities of model parameters desired parameters, both sprung mass acceleration and defined displacement between unsprung mass and tire, calculated.

Performance of semi active suspension under different road disturbances with three amplitude, 0.02, 0.05 and 0.08 [m] on three speed 80, 108 and 130 [km/hr] has been investigated. For modelling and numerical studying suspension parameters of Renault Megane Coupe [10] has been used and they are presented in Table 1. The investigation starts with the car models involving semi active damping with controller and constant stiffness characteristic (see Figure 1).

Table1. Suspension parameters of Renault Megane Coupe [10]

Symbol	Value	Description
m_s	315 kg	Sprung mass
m_{us}	37.5 kg	Unsprung mass
k	29500 N/m	Suspension linearized stiffness
c (u)	500-10000 N.s/m	Suspension linearized damping
k_t	210000 N/m	Tire stiffness

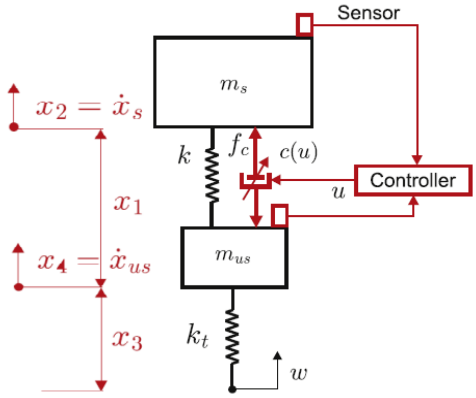

Figure1. Quarter car model [6]

III. MODELLING AND GOVERNING EQUATIONS

Quarter car is a very simple model as it can only represent the bounce motion of chassis and wheel without taking into account pitch or roll vibration modes. However it is very useful for a preliminary design. This model is described by the following system of second-order ordinary differential equations (Figure 1).

In this model x_1 and x_3 are displacements of sprung mass and unsprung mass respectively.

$$m_s\ddot{x}_s = f_c - kx_1 + c(x_4 - x_2) \quad (1)$$

$$m_{us}\ddot{x}_{us} = -f_c - k_t x_3 + kx_1 + c(x_2 - x_4) \quad (2)$$

$$\dot{x}_1 = x_2 - x_4 \quad (3)$$

$$\dot{x}_3 = x_4 - w \quad (4)$$

$$f_c = -c(u)(\dot{x}_s - \dot{x}_{us}) \quad (5)$$

$f(c)$ is the force produced by the damper. Actually, the force in the damper is the product of its damping coefficient and the relative velocity of both ends. $c(u)$ is the damping coefficient, $\dot{x}_s$ is the velocity of the sprung mass and $\dot{x}_{us}$ is the velocity of the wheel. In each vehicle suspension system, there are a variety of parameters which need to be optimized. The trade off between ride comfort and handling characteristics is usually a trial and error procedure which represents an optimization problem. As previously stated, a suspension algorithm is designed to reduce chassis acceleration as well as dynamic tire force. Chassis acceleration is related to comfort and tire force to handling. Dynamic tire force reduction, results in better handling of the vehicle, as the cornering force, tractive and braking efforts developed by the tire are related to normal load, which can be controlled by semi-active methods. Road holding and handling can be quantified by the consideration of the forces and moments applied to the chassis and to the tires. Comfort is more difficult to quantify, although standards exist, the assessment is a controversial issue, because it is an inherently subjective matter.

In this work the best damping coefficient for semi active suspension system, which can produce best ride comfort within great handling has been obtained using numerical simulation. Performance of the semi active suspension has been investigated in three vehicle speed and three road disturbance amplitudes.

IV. RESULTS AND DISCUSSION

For investigating the semi active suspension performance, suspension system has been modelled and simulated using Matlab software with two degrees of freedom. For different damping coefficients, transmitted acceleration to the passenger has been calculated. This is equal to the sprung mass acceleration. For road holding, tire deflection has been used. First, obtained results related to V=80 [km/hr] are presented in Table 2.

Table2. Semi active suspension performance with V=80 [km/hr]

Damping Coefficient [N.s/m]	Disturbance Amplitude [m]	Sprung Mass Acceleration [m/Sec^2]	Tire Deflection [m]
1000	0.02	2.9018	0.0018
	0.05	7.2365	0.0074
	0.08	11.6073	0.0072
2000	0.02	4.1226	0.0037
	0.05	11.3609	0.0148
	0.08	16.4904	0.0146
3000	0.02	5.8737	0.0056
	0.05	15.9508	0.0223
	0.08	23.4947	0.0225
4000	0.02	7.8059	0.0077
	0.05	20.717	0.0297
	0.08	31.2236	0.0308
5000	0.02	9.7165	0.0098
	0.05	25.5398	0.0369
	0.08	38.8659	0.0393
6000	0.02	11.6003	0.0121
	0.05	30.3627	0.0443
	0.08	46.4012	0.0482

For C_s 1000 - 2000 [N.s/m], increasing the road disturbance amplitude from 0.05 to 0.08 [m] increases the transmitted acceleration to the passenger, while tire deflection has been decreased. In other words handling of the vehicle has been improved whereas ride comfort has been decreased. In figures 2 and 3 the effect of increasing amplitude on sprung mass acceleration and tire deflection are shown.

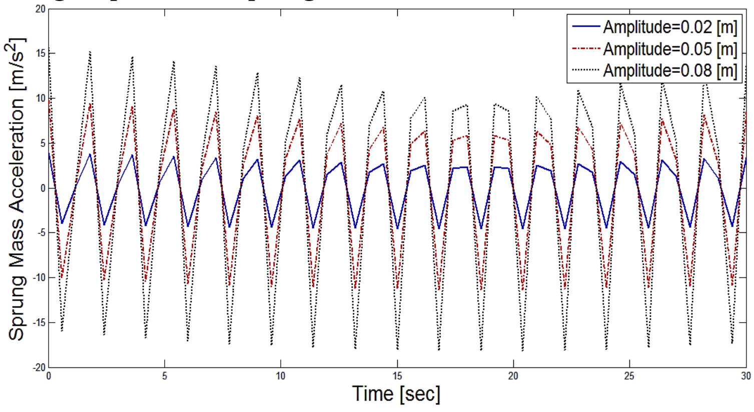

Figure 2. Comparison of amplitude increasing on sprung mass acceleration for Cs=2000 [N.s/m], V=80 [km/hr]

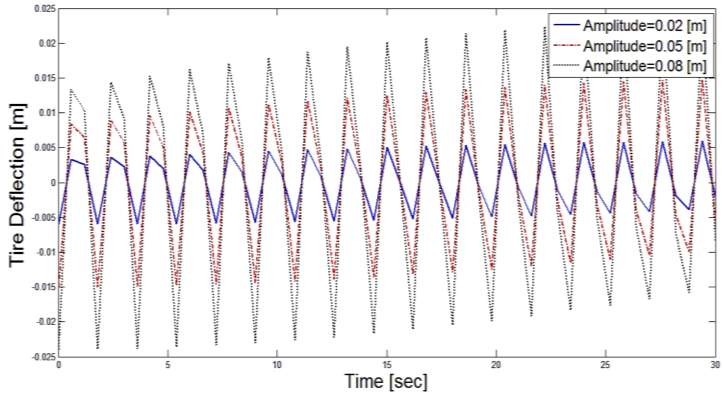

Figure 3. Comparison of amplitude increasing on tire deflection for Cs=2000 [N.s/m] and V=80 [km/hr]

From Table 2, it can be observed that the growth of damping coefficient has direct influence on the sprung mass acceleration. In Table 3, the results for V=108 [km/hr] has been presented. Increasing the amplitude has direct influence on the ride comfort and handling. In Figures 4 and 5 the action of disturbance increasing on spring mass acceleration and tire deflection for *Cs*=2000 [N.s/m], when the vehicle is moving at 108 [km/hr], has been shown. In this case road disturbance amplitudes are 0.02, 0.05 and 0.08 [m].

Table3. Semi active suspension performance with V=108 [km/hr]

Damping Coefficient [N.s/m]	Disturbance Amplitude [m]	Sprung Mass Acceleration [m/sec^2]	Tire Deflection [m]
1000	0.02	3.2511	0.005
	0.05	8.1278	0.0126
	0.08	13	0.0201
2000	0.02	4.8936	0.0076
	0.05	12.2339	0.0189
	0.08	19.5743	0.0303
3000	0.02	6.8263	0.0117
	0.05	17.0658	0.0293
	0.08	27.3052	0.0469
4000	0.02	8.7041	0.0157
	0.05	21.7603	0.0393
	0.08	34.8165	0.0629
5000	0.02	10.6179	0.0191
	0.05	26.5449	0.0478
	0.08	42.4718	0.0764
6000	0.02	12.3682	0.0219
	0.05	30.9206	0.0546
	0.08	49.473	0.0874

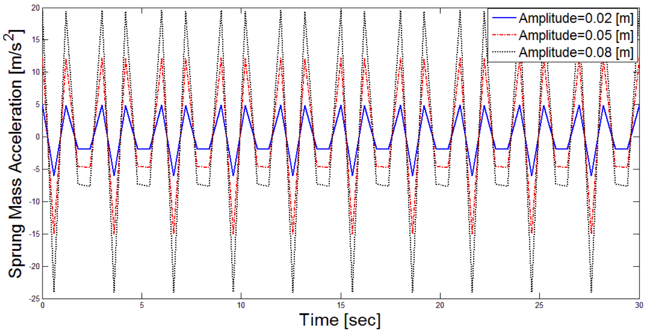

Figure 4. Comparison of amplitude increasing on sprung mass acceleration for Cs=2000 [N.s/m], V=108[km/hr]

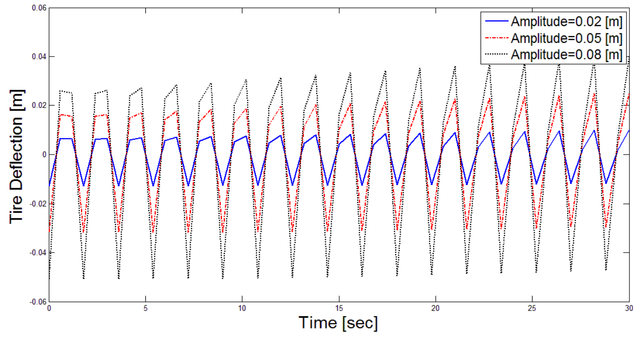

Figure 5. Comparison of amplitude increasing on tire deflection for Cs=2000 [N.s/m] and V=108 [km/hr]

Table 3 shows that the increase of the damping coefficient will enhance the rate of increase for both the sprung mass acceleration and tire deflection. For damping coefficients above 2000 [N.s/m] rate of increase, rises significantly.

Figure 6 and 7 shows effectiveness of ride comfort and handling with increasing road disturbance for Cs=2000 when vehicle is moving at 130 [km/hr].

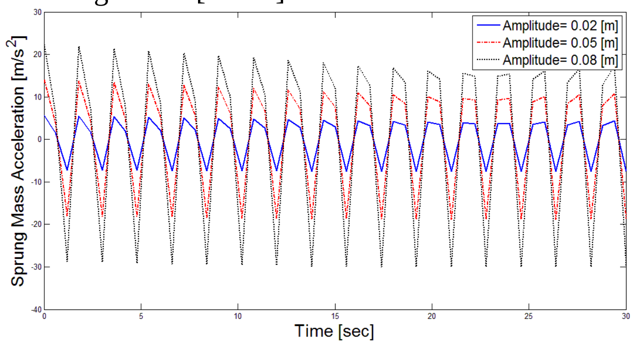

Figure 6. Comparison of amplitude increasing on sprung mass acceleration for Cs=2000 [N.s/m], V=130[km/hr]

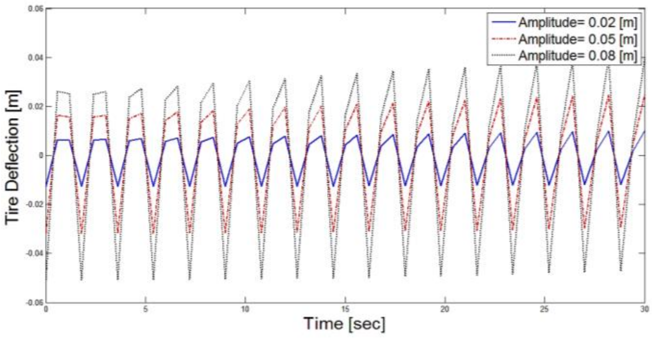

Figure 7. Comparison of amplitude increasing on tire deflection for Cs=2000 [N.s/m] and V=130 [km/hr]

In Table 4, the results are presented for vehicle moving at 130 [km/hr]. For damping coefficient in the range of 1000 to 6000 [N.s/m], sprung mass acceleration and tire deflection have been calculated.

Table 4. Semi active suspension performance with V=130 [km/hr]

Damping Coefficient [N.s/m]	Disturbance Amplitude [m]	Sprung Mass Acceleration [m/sec^2]	Tire Deflection [m]
1000	0.02	2.8583	0.0059
	0.05	7.9822	0.0127
	0.08	12.7716	0.0292
2000	0.02	5.6121	0.0101
	0.05	14.0302	0.0253
	0.08	22.4483	0.0404
3000	0.02	7.0038	0.0155
	0.05	17.5096	0.0388
	0.08	28.0153	0.0620
4000	0.02	8.2638	0.0192
	0.05	20.1160	0.0481
	0.08	33.0552	0.0769
5000	0.02	9.9432	0.0215
	0.05	24.8579	0.0538
	0.08	39.7726	0.0861
6000	0.02	11.3032	0.0229
	0.05	28.2579	0.0577
	0.08	45.2127	0.0915

The obtained results show that increasing the road disturbance amplitude has direct influence either on transmitted acceleration to the passenger or tire deflection. For high value of damping coefficients, the rate of increase will be enhanced. The range of 1000-2000 [N.s/m] for damping coefficient is the most appropriate amount due to its low rate of sprung mass acceleration and tire deflection. For further studying, the performance of semi active suspension system has been investigated in both frequency domain and disturbance amplitude. Frequency range is 0 to 50 [Hz] and disturbance amplitude domain is 0.005 to 0.1 [m]. This will show that how road disturbance amplitude can influence semi active suspension performance in different frequencies.
Figures 8 and 9 show that the frequency range 0-5 [Hz] and disturbance amplitude domain 0-0.06 [m] is the most suitable range for suspension system due to the lowest transmitted acceleration to the passenger.

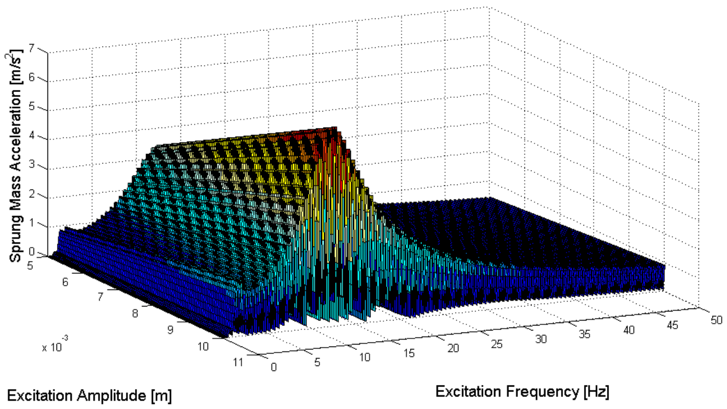

Figure8. Effect of disturbance increasing on sprung mass acceleration for Cs=1000 [N.s/m]

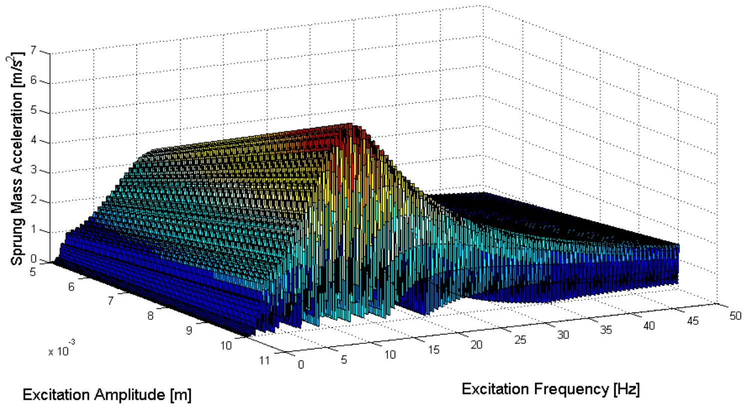

Figure 9. Effect of disturbance increasing on sprung mass acceleration for Cs=2000 [N.s/m]

By increasing the damping coefficient, the rate of acceleration in the range of frequency 0-10 [Hz] and disturbance amplitude domain of 0.08-0.1 [m] have been increased. This can produce great chatter in suspension system. Figures 10-14 show the effect of disturbance increasing on sprung mass acceleration for Cs=3000, Cs=5000, Cs=6000, Cs=8000 and Cs=10000 respectively. It can be observed that for damping coefficients more than 6000 [N.s/m] the rate of acceleration will increase, which has direct influence on decreasing vehicle ride quality and sense of comfortably for passengers.

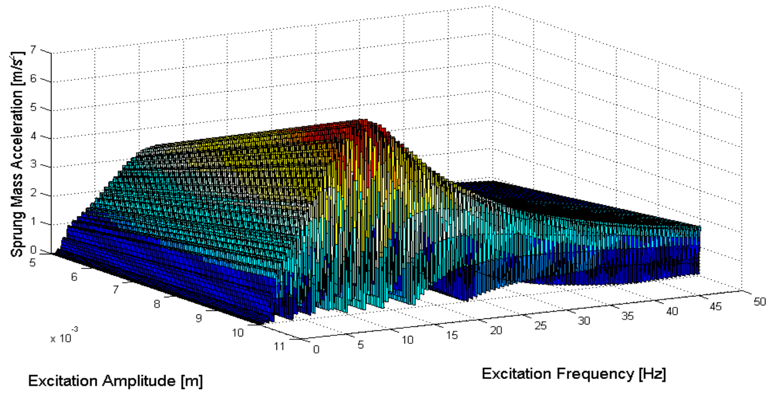

Figure10. Effect of disturbance increasing on sprung mass acceleration for Cs=3000 [N.s/m]

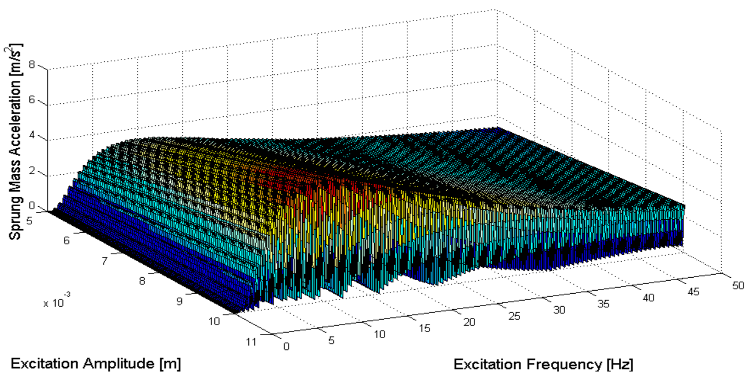

Figure11. Effect of disturbance increasing on sprung mass acceleration for Cs=5000 [N.s/m]

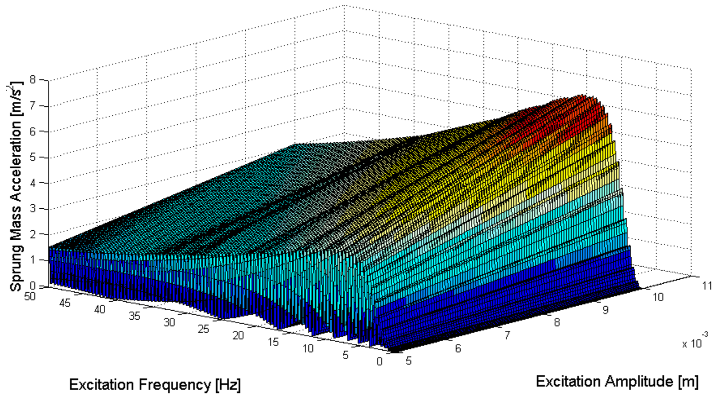

Figure12. Effect of disturbance increasing on sprung mass acceleration for Cs=6000 [N.s/m]

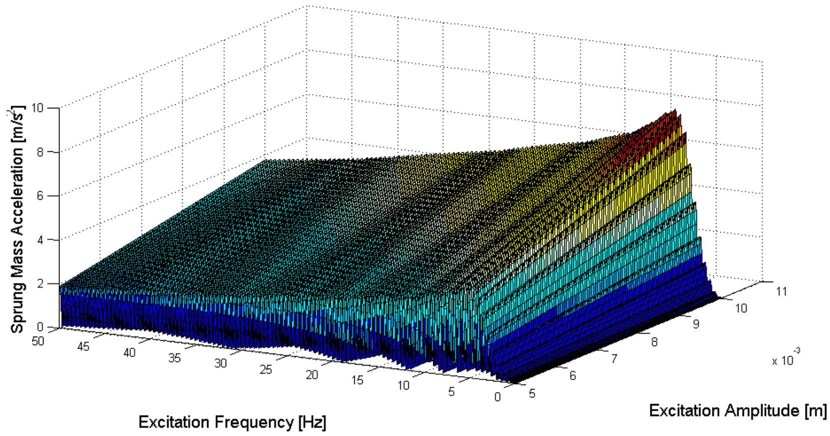

Figure13. Effect of disturbance increasing on sprung mass acceleration for Cs=8000 [N.s/m]

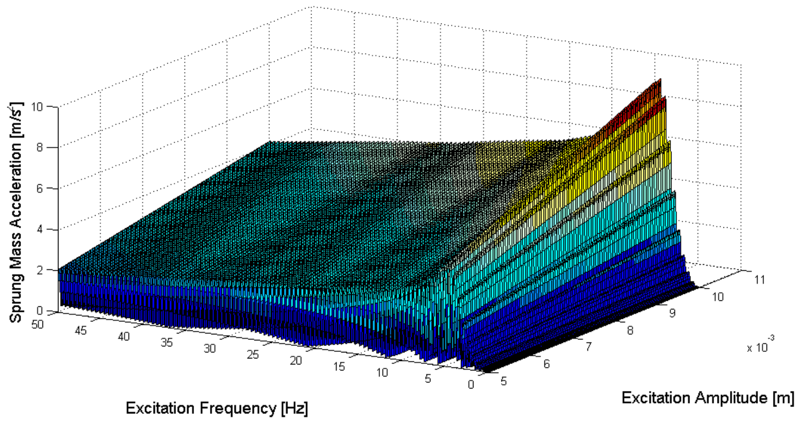

Figure14. Effect of disturbance increasing on sprung mass acceleration for Cs=10000 [N.s/m]

V. CONCLUSIONS

Performance of the designed semi active suspension has been investigated under different road disturbances and different vehicle speeds. The harmonic disturbance amplitudes are 0.02, 0.05 and 0.08 [m] for a vehicle moving at the speeds of 80, 108 and 130 [km/hr]. A quarter car model with 2DOF has been implemented, in addition parameters of Renault Megane Coupe have been used for exact modelling. In different damping coefficients from 1000 to 6000 [N.s/m], ride comfort and handling have been studied. Transmitted acceleration to the passenger has been used for studying the ride comfort parameter and the handling of the vehicle has been studied through tire deflection. The obtained results show that increasing disturbance has direct influence on transmitted acceleration to the passengers which should be controlled. Furthermore, this can increase tire deflection, in other words can reduce handling of the vehicle. From different damping coefficients the range of 1000-2000 [N.s/m] is the most suitable range for Cs, Because it has minimum rate of increase and can reduce tire deflection when vehicle is moving at the speed of 80 [km/hr]. For more investigation, the performance of semi active suspension system has been studied in frequency domain from 0 to 50 [Hz]. In high amount of damping coefficients chatter may occur due to the high rate of increased acceleration The result revealed that the frequencies below 5 [Hz] and the amplitudes less than 0.06 [m] are most appropriate ranges. Subsequently a semi active damper with damping coefficient of 1000-2000 [N.s/m] can produce suitable condition, with low transmitted acceleration to the passenger and therefore good handling of the vehicle.

VI. FUTURE WORK

In this research a quarter car model with 2DOF has been implemented for vehicle modelling, it is worth considering a more complete model with 7DOF for more accurate modelling of passenger car. Moreover; the effect of variation of road disturbance amplitude by respect of time on suspension system may be considered in future studies.

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