ANALYSIS OF PASSIVE AND SEMI ACTIVE CONTROLLED SUSPENSION SYSTEMS FOR RIDE COMFORT IN AN OMNIBUS PASSING OVER A SPEED BUMP

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ABSTRACT

This paper describes the modeling, and testing of skyhook and other semi active suspension control strategies. The control performance of a three-degree-of-freedom quarter car semi active suspension systems is investigated using Matlab/Simulink, model. The objective of this paper is to present a comprehensive analysis of novel hybrid semi-active control algorithms and to compare the semi-active and passive systems in terms of human body vibrational displacements and accelerations. A theoretical model of the human seated model is developed in order to simulate the vertical motion of the Passenger in an omnibus when the vehicle passing over a speed bump. The mathematical model of these systems is presented. Ride comfort of off-road vehicles can be estimated by replacing the normal passive dampers in the vehicle suspension system with controllable, two-state, semi-active dampers.

Key words: *Semi active, Skyhook, Passive, Comfort, Sprung mass, Unsprung mass.*

1. INTRODUCTION

Suspension system design is a challenging task for the automobile designers in view of multiple control parameters, complex (often conflicting) objectives and stochastic disturbances. The roles 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. The purpose of suspension system is to improve the ride comfort, road handling and stability of vehicles.

For vehicle suspension system design, it is always challenging to maintain simultaneously a high standard of ride, handling, and body attitude control under all driving conditions. The problems stem from the wide range of operating conditions created by varying road conditions, vehicle speed, and load. In general, during cornering, braking, and bumping, a high stiffness and damping is needed to provide good handling properties, and to satisfy workspace limitations of the suspension system. However, when a vehicle runs on a low roughness road, a suspension system with low stiffness and damping is needed for good ride comfort. A good suspension system should provide good vibration isolation, i.e. small acceleration of the body mass, and a small "rattle space", which is the maximal allowable relative displacement between the vehicle body and various suspension components [1]. The goal is to simultaneously maintain the suspension travel within the rattle space and to minimize car-body rate-of-change of acceleration.

The vehicle suspension system is responsible for driving comfort and safety as the suspension carries the vehicle-body and transmits all forces between body and road [2].

It is well known that the ride characteristics of passenger vehicles can be characterized by considering the so-called 'quarter-car' model [3]. This method has been widely used to investigate the performance of passive [4], semi-active [5], and fully active [6] suspension systems**.**

Physical models for the investigation of vertical dynamics of suspension systems are most commonly built on the quarter-car model. Greater accuracy is achieved by extensions to a half [7] or full car model [8].

The omnibus passenger seat must be able to isolate the human body from road-induced disturbances. Amongst controlled truck seats a semi-active suspension, usually composed of a controlled damper in parallel with a passive spring, offers a relatively low-cost and reliable solution. A number of control schemes have been proposed for semi-active suspensions over the years [9]. Simulations were found to have very good agreement with experimental data over a wide frequency range (0-20Hz).A detailed measurement of human response to vibration as well as the modeling of the seated human body for the assessment of the vibration experienced was carried out [10].

Semi-active suspensions, which can achieve a ride comfort using less energy than active suspensions have been actively studied during the last decade [11], [12] and [13]. The Sky-Hook Control Law [14] is adapted to many semi-active suspensions.

The skyhook control strategy introduced by Karnopp et al. [15] is the most widely used control policy for semi-active suspension systems. The skyhook control can reduce the resonant peak of the body mass and thus achieve a good ride quality. But, in order to improve both the ride quality and the safety of vehicle, both the resonant peaks of the body and the wheel need to be reduced.

The suspension stroke velocity and vertical velocity of a sprung mass are needed to realize this control law. Generally, these velocities are obtained by differentiating the signals measured with a stroke displacement sensor on suspension systems or integrating the signals measured with an acceleration sensor on a sprung mass.
In the present work a three degree of-freedom quarter car model along with a seat for the semi-active suspension system is analyzed. Different controllers are designed to test the performance of semi active suspension system.

1.1 Suspension system:

Each of the type of suspension has different advantages and disadvantages. Passive vibration control involves an inherent compromise between low-frequency and high-frequency vibration isolation. Passive suspension system consists of an energy dissipating element, which is the damper, and an energy-storing element, which is the spring. Since these two elements can not add energy to the system this kind of suspension systems are called passive.
Passive suspension systems are subject to various tradeoffs when they are excited across a large frequency bandwidth. Compared with passive control, active control can improve the performance over a wide range of frequencies. However, active vibration control has the disadvantages of complexity and high-energy consumption.
Semi-active control has shown many advantages in vehicle suspension systems due to its low energy consumption with similar vibration control performance to the active control methods.
A semi-active control method maintains the reliability of passive control methods and, yet, includes the advantage of the adjustable parameter characteristics of active systems. Semi-active suspension system can offer a compromise between the simplicity of passive systems, and the cost of higher- performance fully active suspension system. In comparison with an active suspension system, a semi-active suspension requires much less power, and is less complex and more reliable and can provide considerable improvement in vehicle ride quality. Consequently, semi-active suspension systems are getting more attention in the development of suspension system.
For a given suspension spring, the better isolation of the sprung mass from road disturbances can be achieved with a soft damping by allowing a larger suspension deflection. However, better road contact can be achieved with a hard damping preventing unnecessary suspension deflections.
The typical transfer-functions of the quarter-car model for the normalized body acceleration and tire load in respect to the road excitation are shown in Fig. 1 [16]. In order to improve the ride quality, it is important to isolate the body, also called sprung mass, from the road disturbances and to decrease the resonance peak of the sprung mass near 1 Hz, which is known to be a sensitive frequency to the human body. In order to improve the ride stability, it is important to keep the tire in contact with the road surface and therefore to decrease the resonance peak near 10 Hz, which is the resonance frequency of the wheel also called unsprung mass.
Therefore, the ride quality and the drive stability are two conflicting criteria. Fig. 2 [16] illustrates this conflict, showing the variation of drive safety and comfort with the changing vehicle parameters body mass, stiffness and damping in the ''conflict diagram''. The conflict diagram presents the vehicles properties, driving comfort and safety for a defined maneuver order to improve the ride stability.

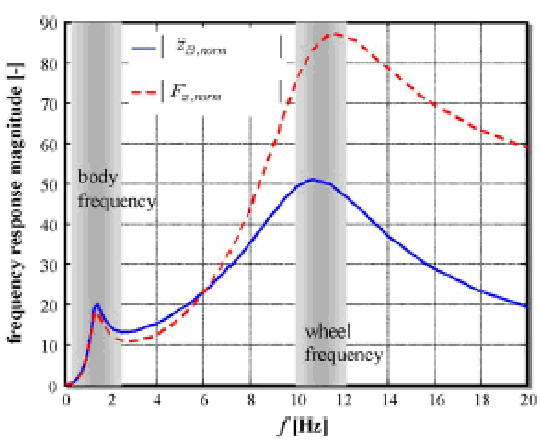

Fig. 1: Frequency response magnitude for normalized body acceleration and tire load for a passive suspension system[16].

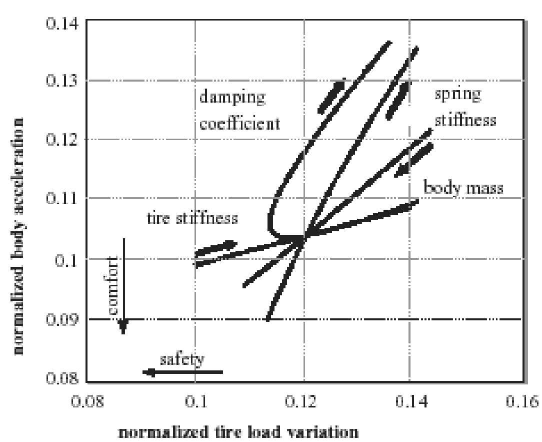

Fig.2: Influence of vehicle parameters, quarter-car simulations[16].

As can be seen from Fig. 2, the fixed setting of a passive suspension system is always a compromise between comfort and safety for any given input set of road conditions on one hand and payload suspension parameters on the other. Semi-active/active suspension systems try to solve or at least reduce this conflict. In this regard, the mechanism of semi-active suspension systems is the adaptation of the damping and/or the stiffness of the spring to the actual demands. Active suspension systems in contrast provide an extra force input in addition to possible existing passive systems and therefore need much more energy. The illustration of Fig. 2 also clarifies the dependency of a vehicle suspension setup on parameter changes as a result of temperature, deflection, and wear and tear. These changes must be taken into account when designing a controller for an active or semi-active suspension to avoid unnecessary performance loss.
Typical features of the different types of suspension are the required energy and the characteristic frequency of the actuator as visualized in Fig. 3 according to [17].

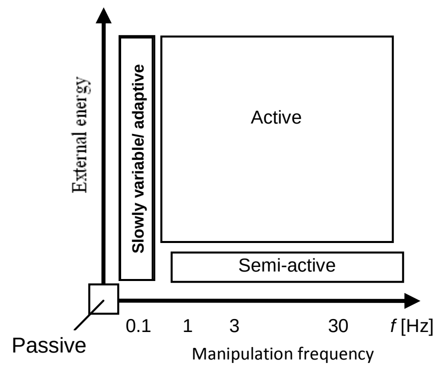

Fig. 3: Comparison between passive, adaptive, semi-active and active systems

To replace complexity and cost while improving ride and handling the concept of semi active suspension has emerged. In this kind of suspension system, the passive suspension spring is retained, while the damping force in the damper can be modulated in accordance with operating conditions. Fig. 4 shows the schematic view of a semi active suspension system along with a seat.
The present work is aimed at an analysis of the effect of suspension on the occupant seat in a vehicle using the quarter car model for a given road input .By using Simulink code, the associated equations of motion are solved to obtain displacements and accelerations on the occupant and to interpret them to evaluate the ride comfort for a specific vehicle. Both skyhook and modified skyhook controls are utilized in the investigations and the results obtained are compared with those obtained for a passive suspension analysis.

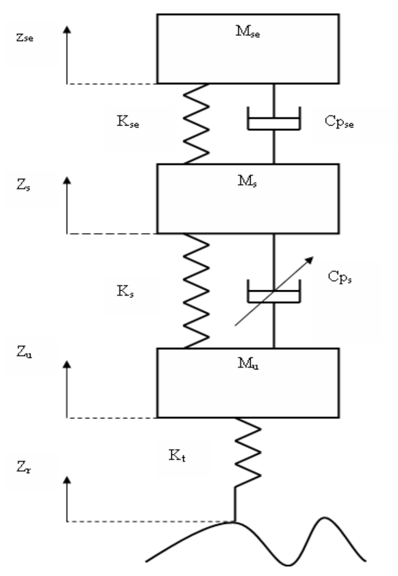

Fig.4. schematic view of a semi active suspension system

2. DYNAMIC MODELS OF A QUARTER CAR:

Physical models for the investigation of vertical dynamics of suspension systems are most commonly built on the quarter-car model.

2.1 Passive system model along with seat:

The piecewise linear model of the passive viscous damper used in the simulation is shown in the Fig.5.

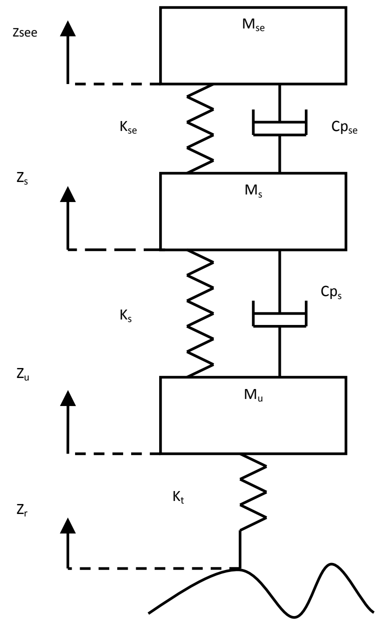

Fig. 5: A quarter car passive model

If Zse is the vertical displacement of the seat, Z_s is the vertical displacement of the sprung mass, Z_u is the vertical displacement of the unsprung mass and Z_r is the road displacement.

The masses of the seat, sprung and unsprung are M_{se}, M_s, and M_u respectively. The corresponding spring stiffness, damping coefficient under the seat are K_{se}, Cp_{se}. Suspension damping and stiffness are indicated as Cp_s and K_s. The tyre stiffness is noted as K_t.

Equation of motion from fig.5 for combined occupant and seat mass is given as

$$M_{se}\ddot{Z}_{se} + K_{se}(Z_{se} - Z_s) + Cp_{se}(\dot{Z}_{se} - \dot{Z}_s) = 0 \quad (1)$$

Equation of motion for sprung mass from fig 5 is

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + Cp_s(\dot{Z}_s - \dot{Z}_u) - K_{se}(Z_{se} - Z_s) - Cp_{se}(\dot{Z}_{se} - \dot{Z}_s) = 0 \quad (2)$$

Similarly, the equation for unsprung mass is

$$M_u\ddot{Z}_u + K_t(Z_u - Z_r) - K_s(Z_s - Z_u) - Cp_s(\dot{Z}_s - \dot{Z}_u) = 0 \quad (3)$$

Let

$X_1 = Z_{se}$, $X_2 = \dot{Z}_{se}$, $X_3 = Z_s$, $X_4 = \dot{Z}_s$, $X_5 = Z_u$, $X_6 = \dot{Z}_u$.

Equations (1),(2) and (3) can be arranged in state space form as

$$\dot{X} = AX + BU$$

$$Y = CX + DU \quad (4)$$

Where $\dot{X}$, Y, A, B, Cand 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. U is an input of the system.
The different matrices are

$$A=\begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ \frac{-K_{se}}{M_{se}} & \frac{-Cp_{se}}{M_{se}} & \frac{K_{se}}{M_{se}} & \frac{Cp_{se}}{M_{se}} & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{K_{se}}{M_s} & \frac{Cp_{se}}{M_s} & -\frac{K_{se}}{M_s}-\frac{K_s}{M_s} & -\frac{C_{se}}{M_s}-\frac{Cp_s}{M_s} & \frac{K_s}{M_s} & \frac{Cp_s}{M_s} \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & \frac{K_s}{M_u} & \frac{Cp_s}{M_u} & -\frac{K_s}{M_u}-\frac{K_t}{M_u} & -\frac{Cp_s}{M_u} \end{bmatrix}$$

$$B=\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \frac{K_t}{M_u} \end{bmatrix} Z_r \quad C=\begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad D=\begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \quad Y=\begin{bmatrix} Zse \\ Zs \\ Zu \end{bmatrix} \quad x=\begin{bmatrix} X_1 \\ X_2 \\ X_3 \\ X_4 \\ X_5 \\ X_6 \end{bmatrix}$$

2.2 Semi-active model:

Semi-active suspension systems are the adaptation of the damping and/or the stiffness of the spring to the actual demands. Figure 6 shows a schematic diagram of a quarter car semi-active suspension control system. The common concept of semi-active springs is based on a system containing an air spring or hydro pneumatic system.

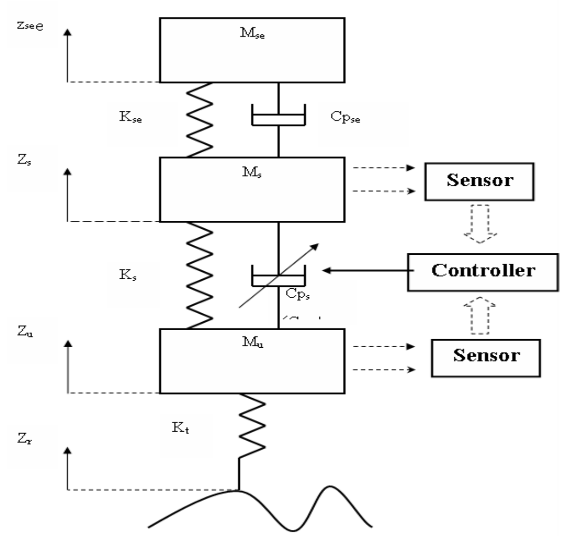

Fig. 6: A quarter car semi-active suspension control system:

3. SEMI ACTIVE CONTROL STRATEGIES:

There are different controls strategies adopted under the semi active suspension system, each one having its own characteristics.

3.1 Limited Relative Displacement Control Method:

The ideal goal of an optimal suspension is to minimize the sprung mass relative displacement and acceleration. However, these two criteria are in conflict. In general, a suspension system with a small relative displacement corresponds to a high sprung mass acceleration, and a large relative displacement corresponds to a low sprung mass acceleration. For this reason, the control strategy is set in a way that the damper is switched to a high damping ratio when the relative displacement is higher than a specific value and a low damping value otherwise. This on–off control law can be expressed as

$$\zeta_s = \begin{cases} \zeta_{max} & |Z_s - Z_u| \geq 0 \\ \zeta_{min} = 0 & |Z_s - Z_u| < 0 \end{cases} \quad (5)$$

Where

ζs is the equivalent damping ratio of the suspension system.

$$\zeta_S = \frac{C_{PS}}{\sqrt[2]{K_S M_S}}$$

This method can limit the relative displacement of the suspension by adjusting two parameters, ζ max and ζ min. This is a simple approach and the results are matching with skyhook control .Therefore they are not presented here. In this paper simulations are run for skyhook (SH) and modified skyhook (MSH) methods only.

3.1 Skyhook Control Method:

Skyhook control is a popular and effective vibration control method because it can dissipate system energy at a high rate.

It is typically classified as continuous skyhook control and on-off skyhook control. The on-off Skyhook controller is usually simpler and better suited for the industrial applications. In this study, on-off skyhook control is implemented. The control law can be described as follows:

This strategy indicates that if the relative velocity of the body with respect to the wheel is in the same direction as that of the body velocity, then a maximum damping force should be applied to reduce the body acceleration. On the other hand, if the two velocities are in the opposite directions, the damping force should be at a minimum to minimize body acceleration. This control strategy requires the measurement of the absolute Velocity of body.

A typical skyhook physical model is shown in Figure 7.

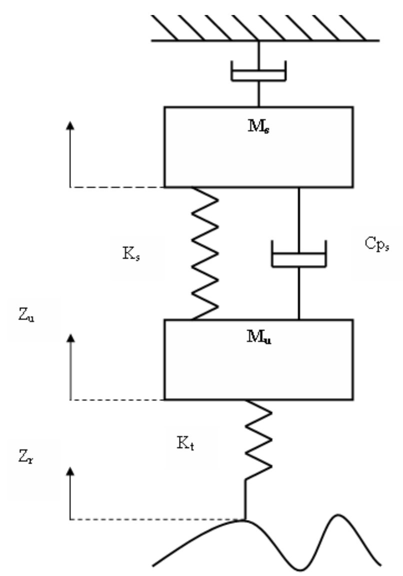

Fig.7: Skyhook control system:

According to the skyhook working principle, the semi-active skyhook control law is

$$\zeta_s = \begin{cases} \zeta_{max} & \dot{Z}_s(\dot{Z}_s - \dot{Z}_u) \geq 0 \\ 0 & \dot{Z}_s(\dot{Z}_s - \dot{Z}_u) < 0 \end{cases} \quad (6)$$

The relative velocity $(\dot{Z}_s - \dot{Z}_u)$ can be obtained by integrating the measured relative acceleration between the sprung mass and the unsprung mass, since the accurate measurement of the absolute vibration velocity of body ($\dot{Zs}$), on a moving vehicle is very difficult to measure.

3.3 Modified Skyhook Control Method:

To overcome the difficulties in the sky hook method this modified method was introduced. In this we can only measure the acceleration of the sprung mass. The absolute sprung mass velocity cannot be estimated in an exponentially stable manner [18]. It could theoretically be obtained by integrating the acceleration of the sprung mass and passing the result through a high-pass filter to remove the direct current (DC) offset. In practice, this is difficult to do because the acceleration offset is not constant and the initial condition of the integral is hard to be determined. For these reasons, the control law (6) is modified to the form (7).

$$\zeta_s = \begin{cases} \zeta_{max} & \dddot{Z}_s(\dot{Z}_s - \dot{Z}_u) \geq 0 \\ 0 & \dddot{Z}_s(\dot{Z}_s - \dot{Z}_u) < 0 \end{cases} \tag{7}$$

where the jerk $\dddot{Z}_s$ of the sprung mass can be obtained by differentiating the filtered acceleration of the sprung mass. Obviously for a sinusoidal input, the phase difference between the jerk $\dddot{Z}_s$ and $\dot{Z}_s$ is π in this case, which is different from the skyhook techniques of the earlier researchers. This control algorithm makes the modified skyhook method easier to implement in a practical system without using a complex observer.

4. IMPLEMENTATION USING MATLAB/SIMULINK:

Figure 8 shows the simulink block diagram for passive suspension system by means of state space approach. The various matrices are entered in to the simulink block and the response for the given input is obtained.

Figure 9 shows the semi active suspension skyhook control and modified skyhook control strategies. Skyhook and modified skyhook controllers are implemented to estimate the passenger comfort when the vehicle passing over the bump with a speed of 40kmph.

In this simulink block diagram an on-off switch is used to actuate control policy. This switch has three input ports which are numbered from top to bottom and one output port.

The first and third input ports are data ports and second input port is control port. As per the control algorithm policy, signal passes through input one when input two satisfies the selected criteria; otherwise it passes through input three. In this way the damper switches back and forth between two possible damping states, high damping and low damping. In this analysis, equivalent damping ratio ζ_s value is varied between 0.1to 0.24 in the first instance and 0 to 0.24 in the second instance. This is to check whether the same results are obtained through adjustment of parameters with in the range of maximum and minimum limits.

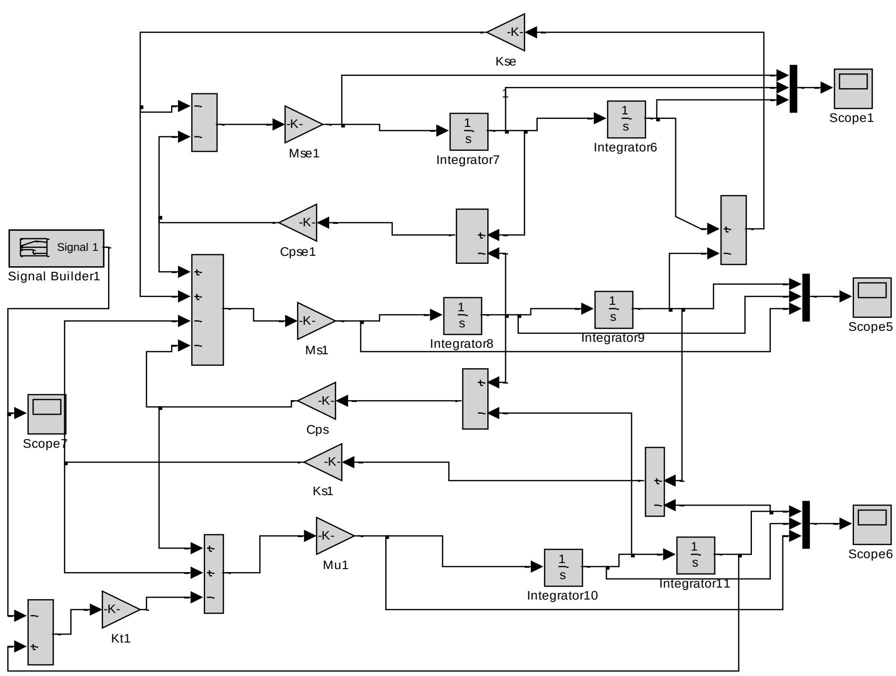

Fig8: Passive simulink block diagram

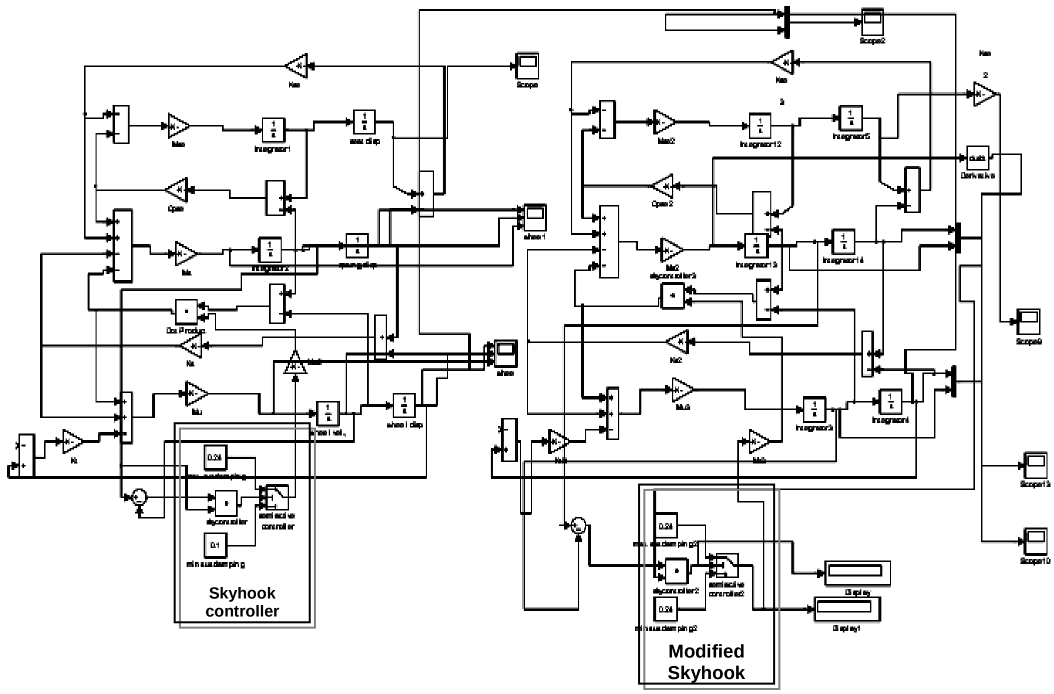

Fig. 9: semi active simulink block diagram

5. SIMULATION RESULT AND DISCUSSION:

Here, Matlab/Simulink is used as a computer aided-control system tool for modeling the non-physical quarter car with its modeling as, all included in one analysis loop passive system, and semi active system. The vehicle parameters considered for the analysis are given in Table 1.For the given in put parameters the response of the system is observed on 10seconds scale. The simulation results for semi active suspension system with modified sky hook control policy show, apparent trade off in between displacement, velocity and acceleration. The semi active suspension system response of the skyhook control peak to peak displacement is less compared to passive system. The seat peak to peak accelerations of the vehicle are increased at the cost of peak to peak reduction of displacements between the passive and semi active.

The important finding for semi active suspension system with modified sky hook control and passive is the response of the both seat and sprung mass dies out faster in semi active system.

Body mass (sprung mass)	8200kg
Mass of the wheel/axle assembly(unsprung mass)	400 kg
Passenger and seat mass	100kg
Suspension damping	5 KNs/m
Suspension stiffness	0.4 MN/m
Passenger seat Damping	6KNs/m
Passenger seat Stiffness	0.1MN/m
Tire stiffness	2 MN/m

Table-1: Parameters of the passenger vehicle

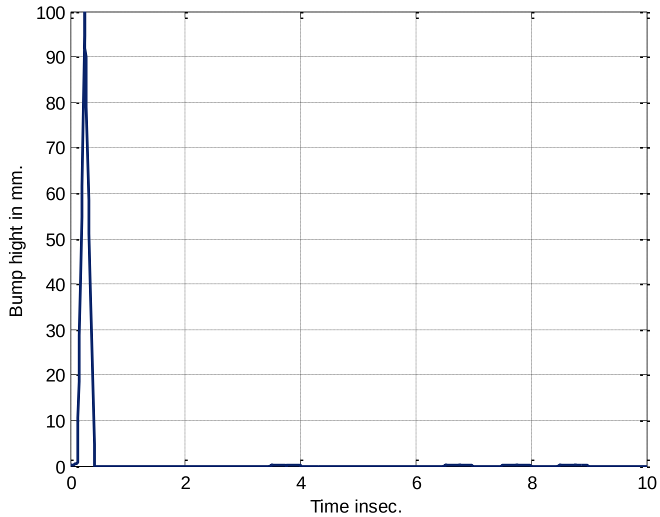

Fig.10: Bump profile (input to the system)

The comparison of all three controllers is presented in Table-2, which shows the peak to peak accelerations and displacements of seat, sprung mass and unsprung mass.

Input	Controller	Max., Seat displacement in mm	Max., sprung-mass Displacement in mm	Max. unsprung-mass displacement in mm	Max., Seat acceleration in mm/sec^2	Max., Sprung-mass acceleration in mm/sec^2	Max.unsprung-mass acceleration in mm/sec^2
Sinu-soidal road input (bump)	Passive	84.11	82.12	92.45	6392	5255	-36250
	SH	69.21	64.56	92.45	6356	4908	-30550
	MSH	68.91	61.21	66.32	2935	3565	-26850
	F.E.M	61.95	55.5	67.52	3187	4152	-31152

Table 2: Maximum values of the time responses of the Quarter car model for vehicle speed of 40kmph

The settling time for the seat under the passive system is 52.94% more as compared to the time for settlement of the semi active system. The sprung mass settling time for passive is 51.13% more as compared to the semi active system. The unsprung mass settling time under the passive is 53.33% more as the semi active controllers.
It may not be out of place here to compare the above results with those obtained from a F.E (finite element) simulation of the omnibus[19] (which is more realistic in view of there being no assumption in respect of ¼ car or ½ car) from the results in Table.2

6. CONCLUSIONS:

The passive and semi-active control methods have been utilized for the analysis of an omnibus. The peak to peak method has been used to investigate the performance of different control methods. The simulation results show considerable differences between the results of passive and different schemes of semi active suspension system. This paper gave an overview of the simulation model which is developed in order to create a research environment for a variety of suspension systems.
After discussing the general conditions for suspension Control and various control concepts, a parameter adaptive suspension control design is presented. In this connection, the potential of improvements for state space feedback in the case of road and body excitation is demonstrated by means of simulations. The variable damper obtains the best results in nearly the same size as an active suspension system.

The current paper has described the performance comparison of passive system, skyhook, and modified sky hook strategies for semi-active suspension systems using computor simulation method. From the results of simulations, it can be stated that the skyhook control can achieve substantial reduction of peak displacement than that of passive suspension.

The results clearly indicate that the modified sky hook method is the optimum robust solution in terms of human comfort. This method is also compared relative to classical skyhook approach improving further the system response in terms of ride.

7. SIMULATION RESPONSES:

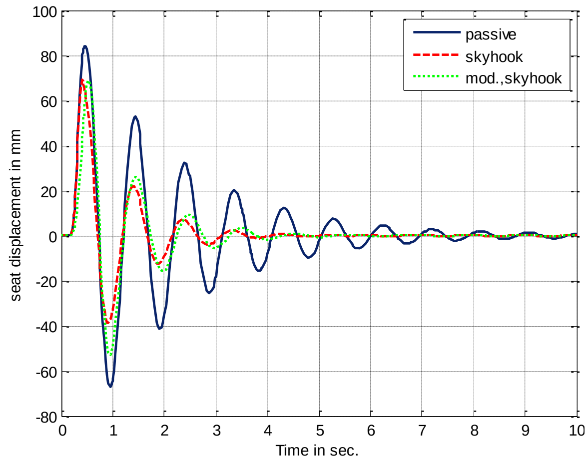

Fig.11: Seat displacement

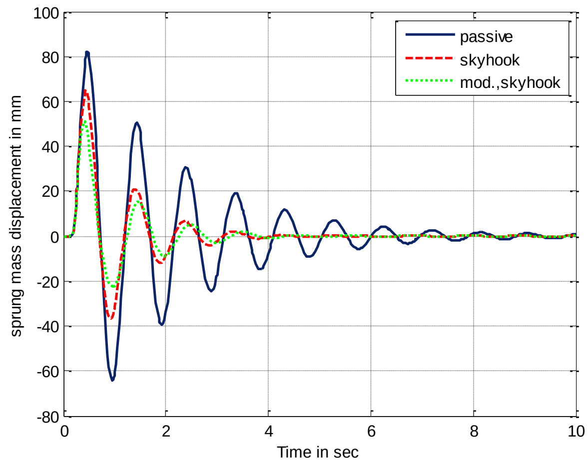

Fig.12: Sprung mass displacement

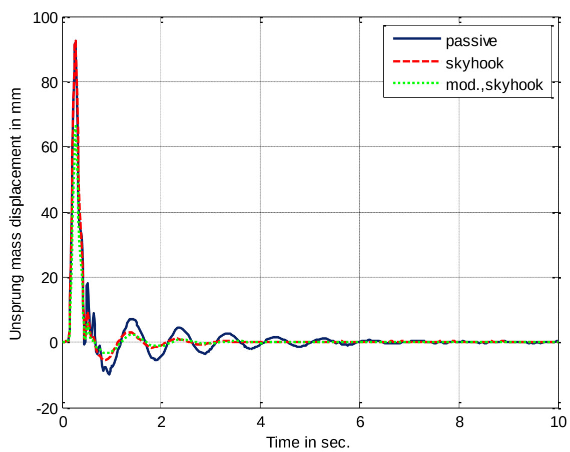

Fig.13: Unsprung mass displacement

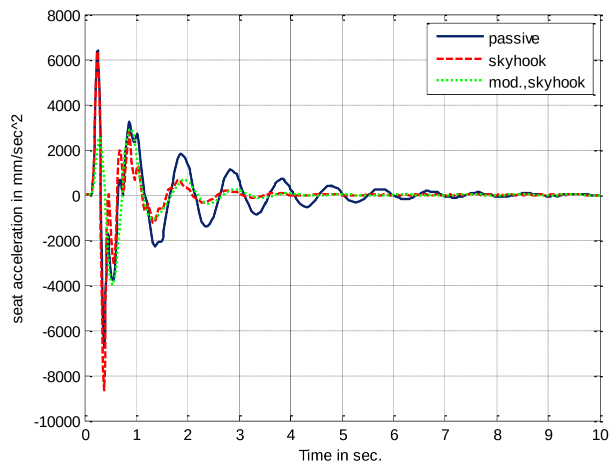

Fig.14: Seat acceleration

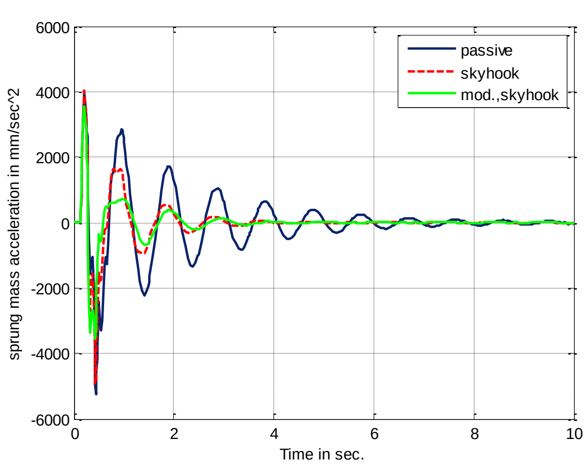

Fig.15: Sprung mass acceleration

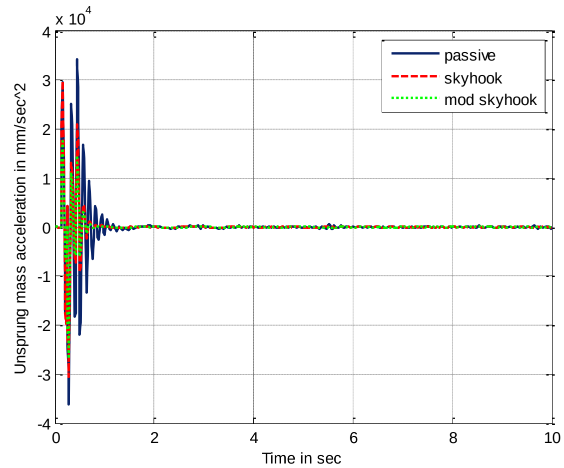

Fig.16: Unsprung mass acceleration

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